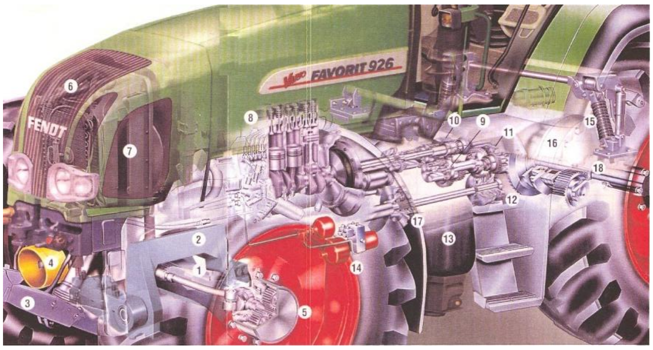
LINKÖPINGS UNIVERSITET

DISEÑO DE UNA TRANSMISIÓN HIDROSTÁTICA



Ignacio Dalda Rivas

Proyecto Fin de Carrera

Director: ***Karl-Erik Rydberg***

Cotutores: ***Vicente Díaz López***

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División de Tecnología de Fluidos

Departamento de Ingeniería Mecánica

Universidad de Linköping, *Suecia, 2009*

RESUMEN DEL PROYECTO

INFORMACIÓN GENERAL

Este Proyecto Fin de Carrera ha sido realizado durante el curso 2008-2009 en la Universidad de Linköping, Suecia, en el departamento de Ingeniería Mecánica de dicha universidad.

El director del proyecto en Suecia ha sido el profesor Karl-Erik Rydberg, especialista en transmisiones hidrostáticas y escritor de varios libros sobre la materia desde distintos puntos de vista. Por otro lado ha habido dos cotutores en la universidad Carlos III del departamento de Ingeniería Mecánica, Vicente Díaz López y María Jesús López Boada. La coordinadora académica en la universidad Carlos III ha sido la profesora Esmeralda Giraldo Casado, del departamento de Organización Industrial.

La redacción y la lectura del proyecto se han realizado en inglés. La fecha de lectura del proyecto fue el día 25 de mayo del año 2009 en la universidad de Linköping ante el profesor Karl-Erik Rydberg junto con Björn Eriksson y Daniel Gunnarsson y la calificación obtenida es de 10 (Matrícula de Honor).

AGRADECIMIENTOS

Dedicado a mis padres, a Rocío, a Carlos y a todas esas personas que han estado ahí siempre que hacía falta.

CONTENIDO

OBJETIVO DEL PROYECTO

Hay diferentes objetivos en este proyecto que han sido analizados convenientemente, no sólo el estudio teórico de las transmisiones hidrostáticas sino también de los diferentes aspectos de la simulación de una transmisión hidrostática. Para ello se ha utilizado el programa de simulación AMESim de manera que se puedan comparar los resultados obtenidos con los que se suponen por la teoría.

En primer lugar la simulación se realiza con el objetivo final de analizar su posible utilización para maquinaria pesada. Este tipo de maquinaria tiene unas características por las que tiene unos altos requerimientos de productividad, capacidad de salida y capacidad general. También es importante saber que por su diseño especial puede ser necesario que el eje de las ruedas tenga la posibilidad de cambiar en gran cantidad su posición y por ello puede ser vital un sistema que pueda transmitir la potencia del motor a través de tuberías flexibles en lugar de a través de los tradicionales ejes.

En la Figura 1 se puede observar una visión general del sistema que se diseña en este proyecto con un motor diesel que envía la potencia a la transmisión hidrostática. Esta transmisión hidrostática consta de una bomba de cilindrada variable y un motor de cilindrada variable (elemento hidrostático primario y secundario respectivamente). La transmisión transmite una velocidad y un par al eje de salida diferente según la cilindrada variable de ambas unidades hidrostáticas. Además el sistema tiene una caja de cambios con al menos dos relaciones para así poder aumentar las posibilidades de potencia transmitida finalmente a las ruedas.

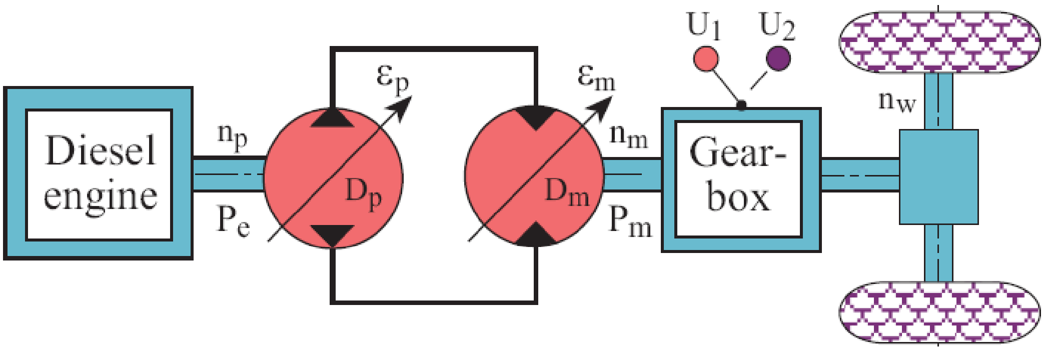


Figura 1: Esquema general de una Transmisión Hidrostática para maquinaria pesada

La forma de diseñar el sistema que se puede ver en la Figura 1 es siguiendo los siguientes pasos.

Lo primero que debe ser diseñado en el programa de simulación es la transmisión hidrostática, empezando con un sistema básico con sólo una bomba y un motor y un sistema de presión constante entre las tuberías de alta presión y las de baja presión para así asegurarse de que la transmisión nunca baja de una presión preestablecida en el lado de baja presión.

Para elegir los parámetros de las unidades hidrostáticas se siguen las características de la transmisión real que se encuentra en el laboratorio de la universidad de Linköping. Algunos ejemplos son por ejemplo la cilindrada máxima de la bomba que es de 36 cm^3 y la del motor que es de 56 cm^3. Otro ejemplo es el volumen de las tuberías de alta y baja presión, que es aproximadamente el mismo en ambos lados e igual a 550 cm^3, es decir 0.5 litros.

Una vez que la transmisión ha sido diseñada con sus características principales es necesario diseñar un sistema para evitar la sobrecarga del sistema debida a la alta presión al mismo tiempo que un sistema de enfriamiento y mantenimiento del lado de baja presión que se encuentran representados en la Figura 2

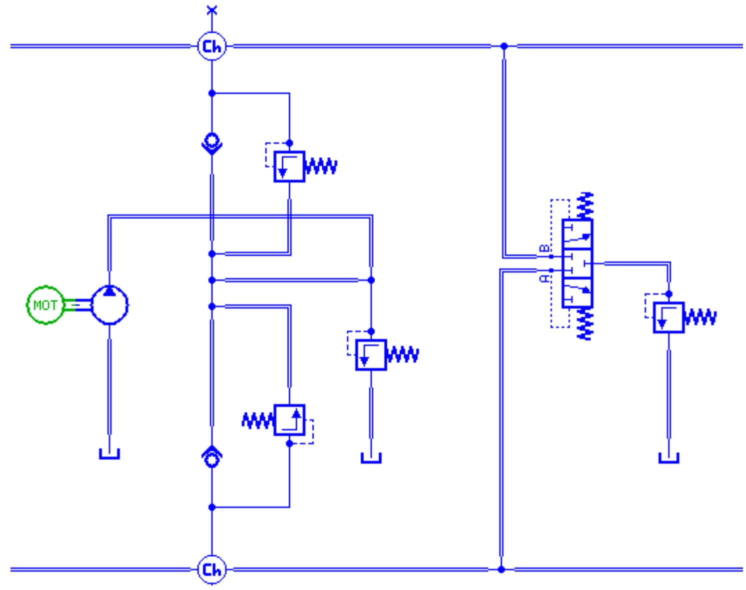


Figura 2: Sistemas de enfriamiento y para evitar sobrepresión y bajadas de presión en las transmisiones hidrostáticas

Una vez que se ha diseñado una transmisión hidrostática muy similar a una real se procede a seguir dos diferentes caminos.

El primero es el diseño de una carga como la que se encuentra en el laboratorio y que consiste en un sistema de válvulas formando un puente de Wheaston que puede funcionar en ambas direcciones y analizar los resultados. (Figura 3)

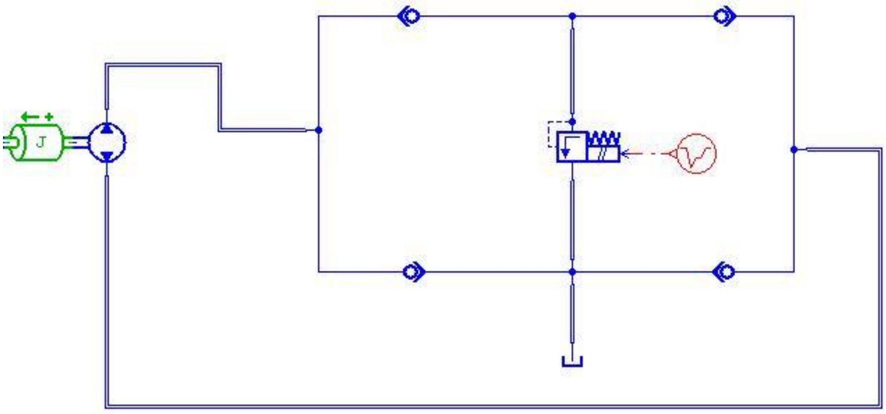


Figura 3: Simulación del sistema del laboratorio (Puente de Wheaston)

El segundo es el diseño de una carga similar a la que se puede encontrar en maquinaria pesada, con una caja de cambios, altas inercias y un gran par debido a las ruedas y a los ejes del vehículo. (Figura 4)

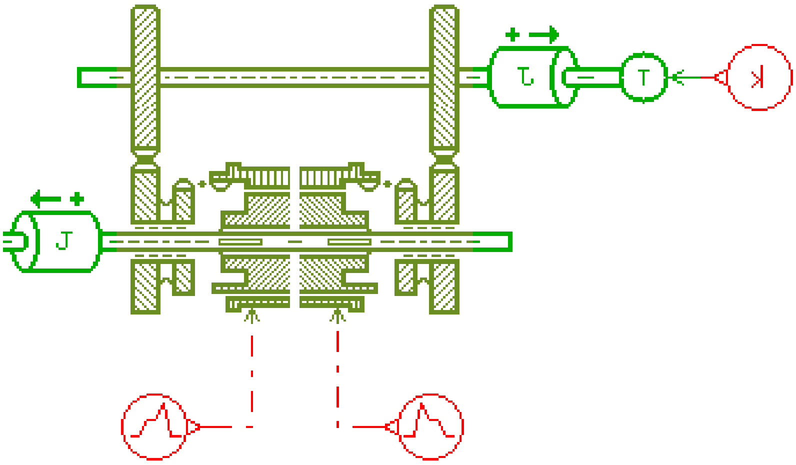


Figura 4: Carga de un vehículo real

Es importante saber que el sistema será cerrado con diferentes filtros para permitir al usuario controlar la velocidad final del eje de salida en vez de tener que controlarlo a través de la cilindrada de la bomba.

Por último se analizarán la frecuencia de resonancia y la amortiguación para diferentes cilindradas del motor y diferentes relaciones de transmisión de la caja de cambios comparando los resultados de la simulación con los esperados en la teoría.

ESTRUCTURA DEL PROYECTO

En primer lugar se han realizado una introducción teórica sobre las transmisiones hidrostáticas y sus relaciones básicas, como puede ser que la velocidad de salida del motor entre la de entrada a la bomba tienen una relación constante e inversa a la que tienen la cilindrada del motor y la bomba.

$$\frac{n_2}{n_1} = \frac{q_1}{q_2} = \varepsilon$$

Posteriormente en un segundo punto se analizan los distintos elementos de una transmisión hidrostática necesarios para producir el movimiento del eje secundario junto con algunos modelos de transmisiones hidrostáticas genéricas. Por último en este apartado se analizan las posibles pérdidas que se pueden producir y que se han de tener en cuenta al realizar los cálculos en las transmisiones hidrostáticas.

El siguiente paso de la teoría es analizar cómo transmite la energía mecánica una transmisión hidrostática y debido a ello las formas de poder variar la velocidad de salida, o la influencia que puede tener una caja de cambios aplicada a una transmisión hidrostática.

El siguiente paso es el diseño de la transmisión hidrostática mediante el uso del programa de simulación AMESim para poder hacer las diferentes pruebas necesarias. Es muy importante que dicho diseño además de basarse en la transmisión que estaba disponible en el laboratorio de la universidad de Linköping generara resultados robustos y fiables.

Debido a los objetivos del proyecto lo siguiente es la generación de una carga aplicada a la salida del motor de la transmisión de dos maneras distintas, por un lado una carga como la que tenía la transmisión del laboratorio y por otro lado una carga que simula un vehículo real del estilo a los que podrían requerir el uso de una transmisión como esta, como pudiera ser una máquina agrícola o similar.

El siguiente paso es el análisis de los resultados que se obtienen utilizando las diferentes relaciones de la caja de cambios disponible en el sistema con una carga similar a la de un vehículo real.

Finalmente se pasa a realizar el análisis dinámico de la transmisión hidrostática con las diferentes cargas diseñadas comparando los resultados obtenidos con los previstos teóricamente.

RESULTADOS IMPORTANTES

En el análisis dinámico de la transmisión hidrostática se realizan distintos estudios con y sin realimentación del sistema en los que se analizan esencialmente los datos a la salida del motor de la transmisión para poder comprobar los datos de la resonancia resultante y del amortiguamiento.

En primer lugar se decide introducir una señal de entrada de forma que tenga un salto instantáneo y así poder observar cómo reacciona el sistema a dicho salto. En la Figura 5 se puede ver la señal de entrada en color rojo, la cual sube lentamente hasta 0,4 y se estabiliza hasta que en una décima de segundo pasa a tener un valor de 0,6. Esta señal se corresponde con el tanto por uno de la cilindrada de la bomba de la transmisión hidrostática pero la señal verde corresponde con la realimentación que es la señal que finalmente se envía a la bomba de forma que se evitan las sobrecargas en el sistema.

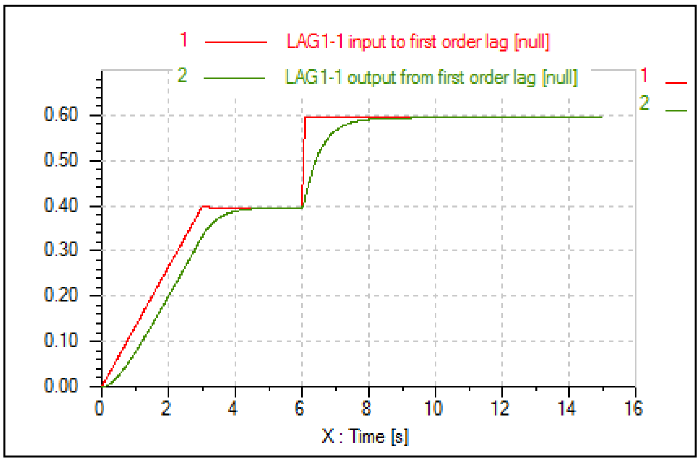
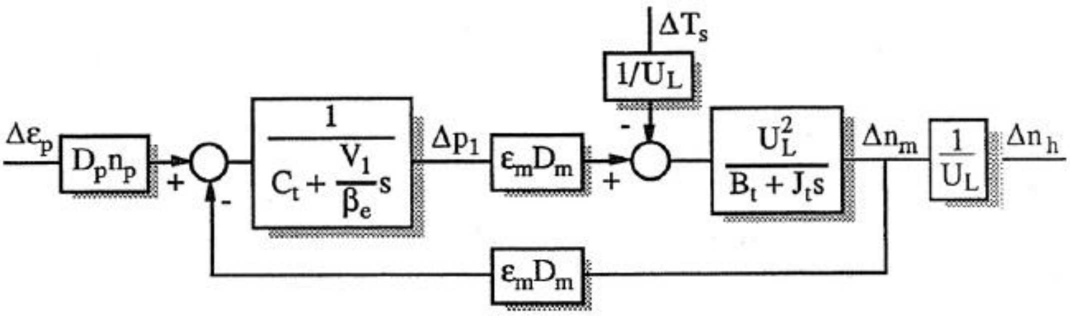


Figura 5: Señal de entrada antes y después de la realimentación

Una vez introducida la señal de entrada y mediante diversos análisis de la función de transferencia que se obtiene del sistema:



$$\frac{\Delta\varepsilon_p}{\Delta i_{ps}} = K_{ps}G_{ps} = K_{ps}\frac{1}{1+\dfrac{s}{\omega_{ps}}}$$

A partir de la función de transferencia y variando poco a poco las relaciones de transmisión de la caja de cambios que está situada entre la transmisión hidrostática y las ruedas del vehículo se pueden observar los rangos de la frecuencia de resonancia y del amortiguamiento que han de seguir tendencias opuestas y lineales.

$$\omega_{hmin} \leq \omega_h \leq \omega_{hmax}$$

$$\delta_{hmax} \geq \delta_h \geq \delta_{hmin}$$

Por último el resultado entre el amortiguamiento $\boldsymbol{\delta_h}$ y la frecuencia de resonancia $\boldsymbol{\omega_h}$ ha de ser un valor constante. A continuación se pueden ver los resultados obtenidos en la Figura 6.

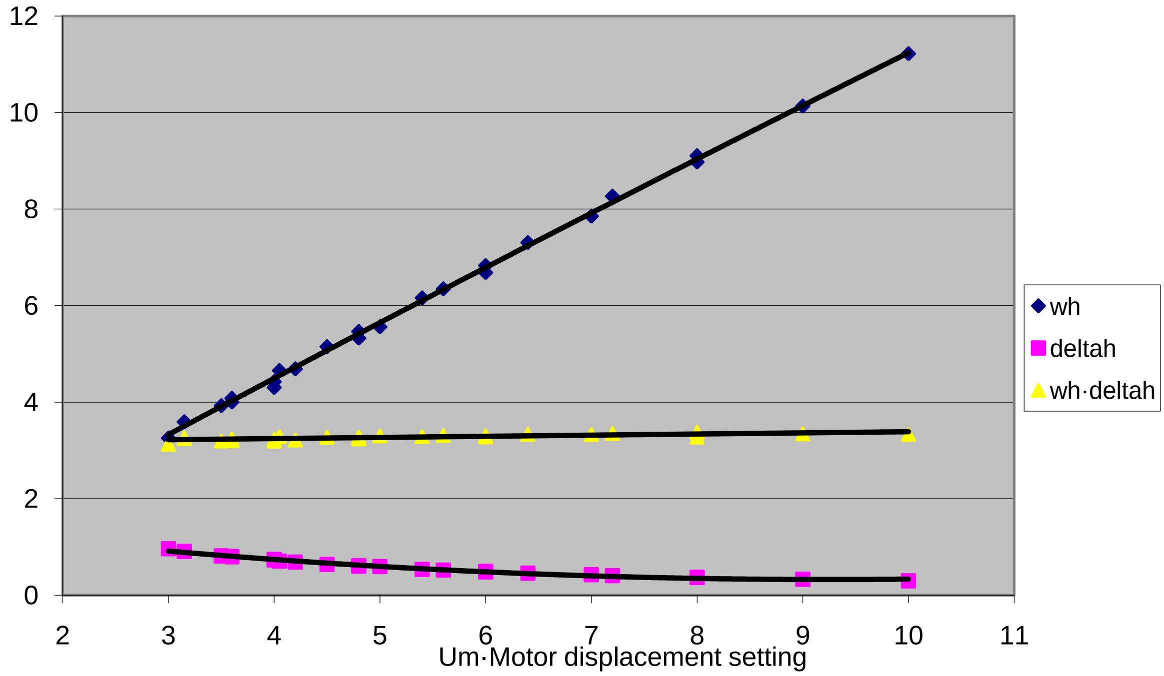


Figura 6: Resultados del análisis dinámico de la transmisión hidrostática

Como se puede observar el resultado no es exactamente como se puede esperar teóricamente, pues el amortiguamiento no decrece de forma lineal y por tanto el producto $\boldsymbol{\omega_h \delta_h}$ tiene una ligera tendencia ascendente.

Las razones para entender esto son que cuando la caja de cambios tiene una diferente relación de transmisión transmite una inercia distinta a la transmisión hidrostática y cuanto mayor es la inercia mayor es el amortiguamiento. La segunda razón para esta tendencia es que la caja de cambios tiene su propio amortiguamiento implícito pero que no crece linealmente al cambiar la relación de transmisión.

CONCLUSIÓN

En este proyecto han sido analizados algunos aspectos de las transmisiones hidrostáticas para posteriormente, a partir de dicha información, diseñar una buena transmisión hidrostática para simular diferentes situaciones que podían ser interesantes para este tipo de sistemas.

Después de todo, la transmisión diseñada e implementada con AMESim ha demostrado su robusto diseño gracias a los sistemas que posee de amortiguamiento y a las medidas de seguridad ante la sobrecarga y el calentamiento.

Se ha observado que con la carga similar al laboratorio, compuesta por un puente de Wheaston, el sistema posee un gran amortiguamiento. Por ello el sistema es capaz de absorber rápidos cambios en la señal de entrada sin problemas de sobrecargas, o largas frecuencias de resonancia. Por ello se demuestra que la carga elegida para el laboratorio de la universidad de Linköping soportaría los análisis reales.

La segunda carga diseñada para vehículos reales del estilo maquinaria pesada o agrícola no tiene un amortiguamiento tan grande. Por ello, y para solucionar posibles problemas de sobrecarga lo mejor es utilizar un filtro paso bajo que reduce la velocidad de los cambios a la entrada. En este diseño también es necesario el uso de un controlador PID para poder manejar directamente la velocidad de salida en vez de la cilindrada variable de las unidades hidrostáticas.

Todos estos sistemas han demostrado que generan un sistema más suave en su funcionamiento y utilizable.

Desde otro punto de vista, este tipo de transmisiones han demostrado una alta controlabilidad, especialmente en bajos rangos de velocidad, lo cual es fundamental para su aplicación en maquinaria pesada. También es importante que las transmisiones hidrostáticas son capaces de operar con altos pares y permitiendo potencias del orden de 200kW.

A raíz de los análisis realizados, los resultados demuestran que una correcta combinación entre la principal transmisión hidrostática la caja de cambios y el sistema de control es posible alcanzar una gran eficiencia en un gran rango de utilización.

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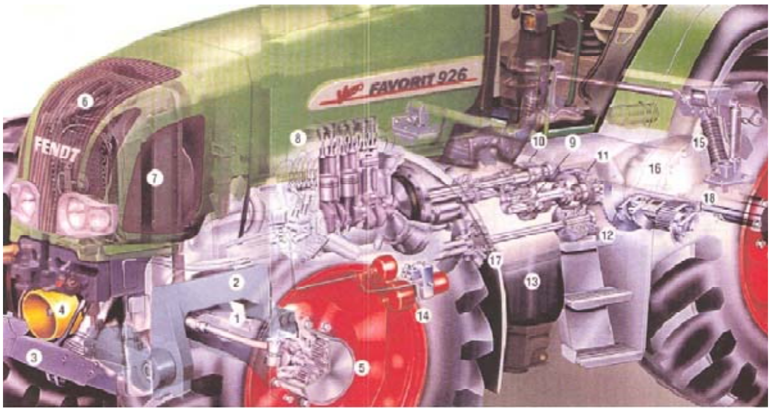
HYDROSTATIC TRANSMISSION DESIGN



Ignacio Dalda Rivas

Master Thesis in Hydrostatic Transmissions TGZD20

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Linköping Universitet, Sweden, 2009

ABSTRACT

This project treats the analysis of the hydrostatic transmission dynamic properties using a simulation model, which has been done with the simulation program AMESim. That simulation has been the main work in the project, especially because it is a good way to understand how a hydrostatic transmission works.

The hydrostatic transmission are used in heavy vehicles such as earth moving machines, agriculture machines, forest machines, industrial and mining lifters. Nowadays, the demand of that kind of transmissions is increasing because of hydraulic drives have many advantages over other technologies. That is because of hydrostatic transmission has a high output capacity combined with high overall efficiency over a wide velocity range, and all that, with a low weight and volume. So, the first step was to develop a simulation for a transmission used in heavy machinery. Thus, that model has a gearbox, high inertias and high torque corresponding to the wheels and the shafts of the vehicle. Also a simulation of a transmission located in a test stand has been done. Next to that, the efficiency has been improved in a wide used range. To do that, a control system for the transmission pump and motor has been implemented. With it, the controllability of the system is much better in different loading conditions. And also with all this information, the objective is to be able to choose the main transmission components (pump, motor and gearbox).

The real transmission which is in a test stand in the Linköping University laboratories has been taken as a model to set the parameters of this simulation model.

Another important thing has been the validation of the simulation model by comparing its results with the theoretical results, which has been calculated using the hydraulic equations.

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1. INTRODUCTION

The objective of this project is the design of a Hydrostatic Transmission thinking on heavy vehicles like earth moving machines, or agriculture machines because of their special requirements.

Now the principal purpose is to resume the main characteristics of Hydrostatic Transmissions and how do they work.

The mechanical energy of the input shaft is transformed in a way of pressure energy of an incompressive liquid, and after is transformed again to mechanical energy in the output shaft.

The main parts are the pump, drove by the "input shaft" which sends the pressure liquid to a motor connected to the "output shaft".

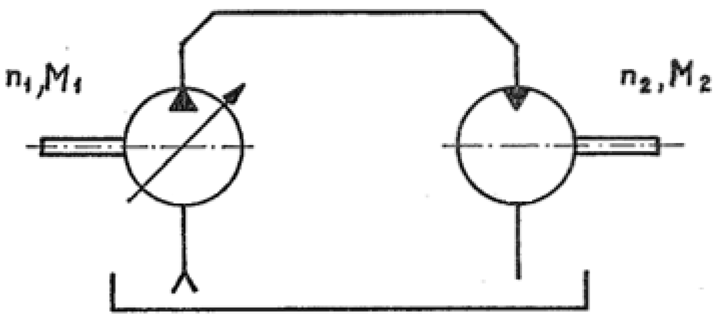

Figure 1-1: Hydrostatic transmission in an open circuit with variable pump [1]

The primary hydrostatic unit is the one connected with the input shaft while the secondary hydrostatic unit is connected with the output shaft (simplifying they can be called primary and secondary).

If the picture of a hydrostatic unit has an arrow like the primary in Figure 1-1 means that it has a variable displacement volume, which is the flow for an input constant speed depends on the actuation on the displacement volume. This flow makes a proportional speed on the output shaft.

Without the logical losses it can be said that the relation between the input speed (n_1) and output speed (n_2) is the same than the effective displacement volume relation.

$$\frac{n_2}{n_1} = \frac{q_1}{q_2} = \varepsilon \tag{1-1}$$

The relation between the input and the output torque is the inverse of the speed relation (without losses)

$$\frac{M_2}{M_1} = \frac{q_2}{q_1} = \frac{1}{\varepsilon} \tag{1-2}$$

So the product of speed and the torque is proportional to the transmitted power.

The main difference between the hydrostatic transmission and the hydrodynamic transmissions is that the first group transmits the power by the flow static pressure while the second group transmits the power by kinetic energy due to the liquid speed.

So in the hydrostatic transmissions, the static pressure is much bigger than the dynamic one because the speed of the liquid is low.

To obtain a high power density with low sizes are necessary relatively high operation pressures. These pressures also need to be during a long time, so the resistance to high pressures along the time is one of the main characteristics to take into account. But also is very important the size of the transmissions because a smaller one needs less space and less weight and then can be positioned easier in the vehicle. That is the reason why the flow speed has to be as high as possible (is important to be careful with the strangling on the sliding parts), and as small as the transmission is, the flow speed must be higher.

There are different ways to make the hydrostatic transmissions. They can be all the parts into one block, but if it is required by the conditions of the machine which is going to be used it is possible to separate the primary and the secondary part, connecting them by pipes. This possibility makes the hydrostatic transmissions really adaptable to all kind of machinery.

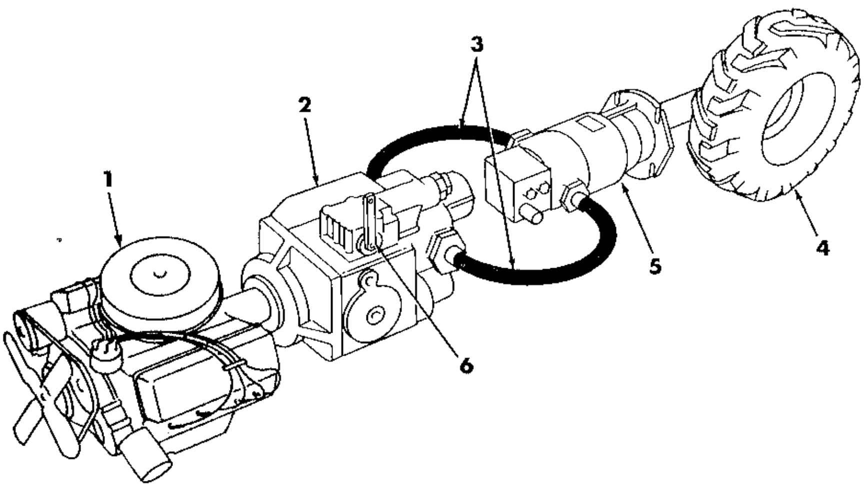

Figure 1-2: Hydrostatic transmission with primary and secondary connected only by pipes [17]

The fluids used in hydrostatic transmissions have to be with low compressibility because of the use of pressure or liquid circulation energy, but also have lubrication capacity for all the hydrostatic parts. Also the fluids have to have the ideal viscosity which avoids internal leakage flows but also they can't have important temperature changes. These required characteristics made the first hydrostatic transmissions the use of lubrication oil but nowadays the technical has created specific hydraulic oils which can work better. These fluids have to be completely isolated from the atmosphere because any contact between the turbulence fluid and the air will increase the compressibility too much creating function troubles.

The principal advantages of the hydrostatic transmissions are the capacity of changing the output speed easily, but also keep it in spite of the variations on the output torque if the input speed keeps constant, the possibility of divide the transmission in two parts, and also the easiness to reverse the turn direction. But in other side the principal limitation of the double change between mechanical energy to pressure energy and then again to mechanical, with the logical power losses and cost of these losses.

2. HYDROSTATIC POWER PRODUCTION

2.1 Hydrostatic elements

The hydrostatic power transmission is carried out by the conversion of mechanical energy in the liquid flow over high pressure. This conversion is executed by the hydrostatic units (also called pump and hydraulic motor).

The hydrostatic units are volumetric or displacement energy converters. When the pump is running it has two different phases, admission and trust. To describe a completely cycle, the liquid fill a volume (usually the cylinder), then the suction pipe is insulated for the reduction of its volume until the dead centre while the thrust side is opened sending the liquid under high pressure. Once this point is achieved the suction side is opened again and also the thrust side closed for the beginning of the cycle again. This process is carried out also by the motor in the opposite direction.

In a hydrostatic unit which can only be used as a pump there will be non return valves on the admission and thrust pipes (usual behaviour). These valves act with the liquid pressure to establish the correct communications at the correct moment. But also they guarantee the exact distribution with low velocities, while with high velocities the mobile part inertias cause on these pieces delays in the beginning.

In a hydrostatic unit which can only be used as a motor the liquid distribution has to be forced with distribution piston rods or valves active by the eccentric assembled over the element shaft.

An element which can be used as a pump or a motor indifferently will be called transmission hydrostatic unit.

The most important magnitude for the hydrostatic units is the displacement volume, which is the quantity of liquid pushed per one revolution of the shaft with low pressure (D).

The final objective of a hydrostatic transmission is to obtain variable speed at the output shaft while the speed at the input shaft is constant. For do it the displacement volume of the hydrostatic units has to be changed. It can be done regulating the elements slope if it is an axial pistons unit or changing the elements eccentricity if it is a radial piston unit.

In general, the way to quantify this variation is the displacement setting $\boldsymbol{\varepsilon}$ which is a value between 0 and 1 (from no displacement volume to its maximum value) and also from 0 to -1 if the hydrostatic unit can have an inverse flow.

The rotation velocity of a hydrostatic unit **n** can be given in revolutions per minute or in radians per second, but the first one is used in the simulation program AMESim so then will be used in this project.

To obtain the flow without taking in account the losses it has to be used the First Fundamental Equation:

$$q = nD = n\varepsilon D_0 \tag{2-1}$$

Where Q is the flow and D_0 is the maximum displacement volume of the hydrostatic unit.

The Second Fundamental Equation, without taking into account the torque losses gives the relation between the pressure and the torque:

$$p = \frac{M}{D} = \frac{M}{\varepsilon D_0} \tag{2-2}$$

Finally, the Third Fundamental Equation gives the different relations to obtain the power:

$$P = qp = nM = pn\varepsilon D_0 \tag{2-3}$$

An easy example to see the difference between a pump and a motor is a hydrostatic unit which works under constant torque and pressure. Looking for the second fundamental equation, if the torque is bigger than the pressure the unit will work as a pump, while in the opposite case it will work as a motor.

There are many ways to change the displacement volume of a hydrostatic unit but the basic idea is to change the piston displacement. The Figure 2-1 shows an example of a variable displacement pump and a variable displacement motor.

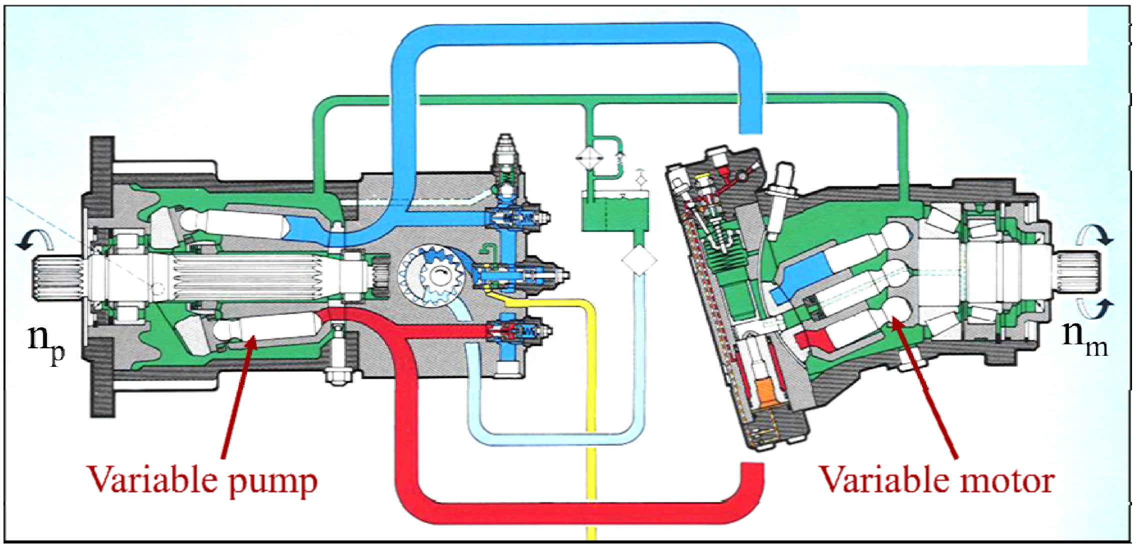

Figure 2-1: Variable displacement pump and motor [11]

2.2 Hydrostatic units with more than one cylinder

There are some different ways to put the pistons, like a radial distribution as in Figure 2-2, or with axial pistons as in Figure 2-3, or also with axial pistons but with swivelling head for change the displacement volume changing the slope as in Figure 2-4.

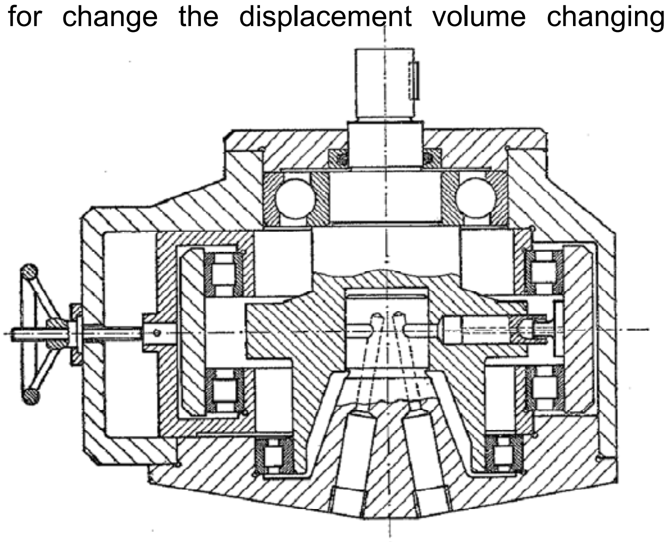

Figure 2-2: Hydrostatic unit with radial pistons [1]

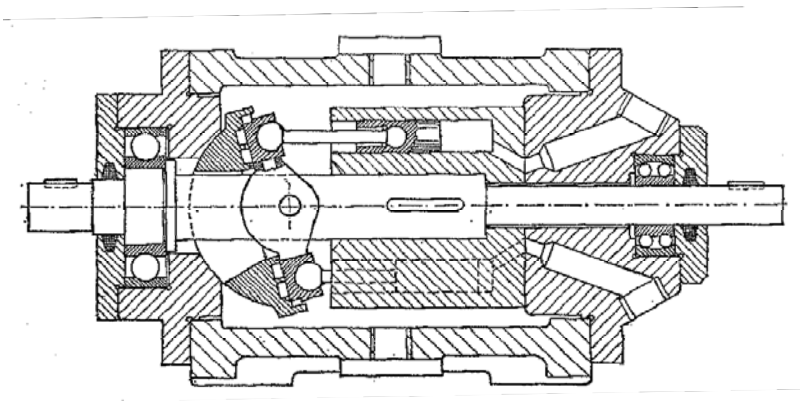

Figure 2-3: Hydrostatic unit with axial pistons [1]

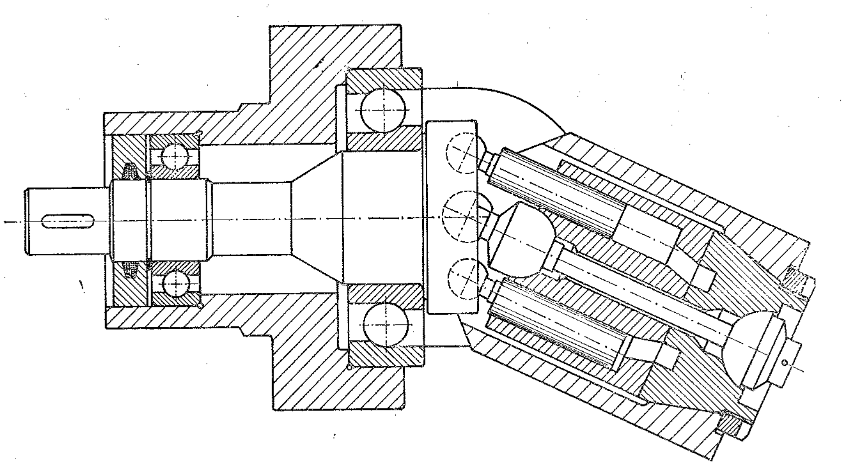

Figure 2-4: Hydrostatic unit with axial pistons and swivelling head [1]

2.3 Hydrostatic Unit losses

In this point are going to be analyzed the main losses, which are due especially to leaks. Nevertheless, there are also other kinds of losses due to dry and viscous friction or due to the hydrodynamic losses.

At first it is shown the effective flow for the pump and motor, where η_{vol} is the volumetric efficiency.

$$q_{ep} = \varepsilon_p D_p n_p \eta_{volp} \tag{2-4}$$

$$q_{em} = \varepsilon_m D_m n_m \eta_{volm} \tag{2-5}$$

It is also important to know the influence of the losses in the torque, which is included by the volumetric efficiency.

$$M_{in} = \frac{\varepsilon_p D_p}{2\pi} \Delta_p \frac{1}{\eta_{hmp}} \tag{2-6}$$

$$M_{out} = \frac{\varepsilon_m D_m}{2\pi} \Delta_m \frac{1}{\eta_{hmm}} \tag{2-7}$$

As the leakage flow is determined by the next equation.

$$q_l = C_v \frac{\Delta_p}{|\varepsilon_p| n_p \eta} \tag{2-8}$$

Then, the volumetric efficiency can be calculated as it is shown.

$$\eta_{volp} = 1 - C_v \frac{\Delta_p}{|\varepsilon_p| n_p \eta} \tag{2-9}$$

$$\eta_{volm} = 1 - C_v \frac{\Delta_p}{|\varepsilon_m| n_m \eta} \tag{2-10}$$

To calculate the volumetric efficiency only the C_v (laminar leakage losses) is used. Nevertheless, there is another efficiency which takes into account the other looses also. This is the hydraulic mechanical efficiency.

$$\eta_{hmp} = \frac{1}{1+\left(k_p + k_v \frac{n_p \eta}{\Delta_p}\right) e^{\left(k_\epsilon (1-|\varepsilon_p|)\right)}} \tag{2-11}$$

$$\eta_{hmm} = \frac{1}{1+\left(k_p + k_v \frac{n_m \eta}{\Delta_p}\right) e^{\left(k_\epsilon (1-|\varepsilon_m|)\right)}} \tag{2-12}$$

But is usual to avoid the using of those equations for then use some reference equations to include the losses. A good example is in the Figure 2-5 where the torque losses are represented by a disk brake and the leakage flow with a fixed hydraulic orifice between the high and the low pressure side.

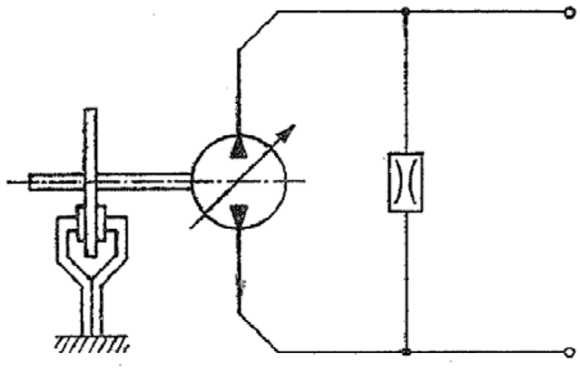

Figure 2-5: Hydrostatic unit with a break and a fixed hydraulic orifice for simulate the losses [1]

3. TRANSMISSION AND CONTROL OF MECHANICAL POWER

3.1 Control of variations in output speed

It deals about power transmission with a variable shaft speed. It is considered that the transmission has a constant input speed, and a variable output speed, which is a control signal function.

In a first approximation, the diagram has an engine, which is coupled to the input shaft of the variable transmission, whose output shaft is connected to a load. The output speed is determined by the position of the control lever of the transmission, but the output torque will also have a certain influence.

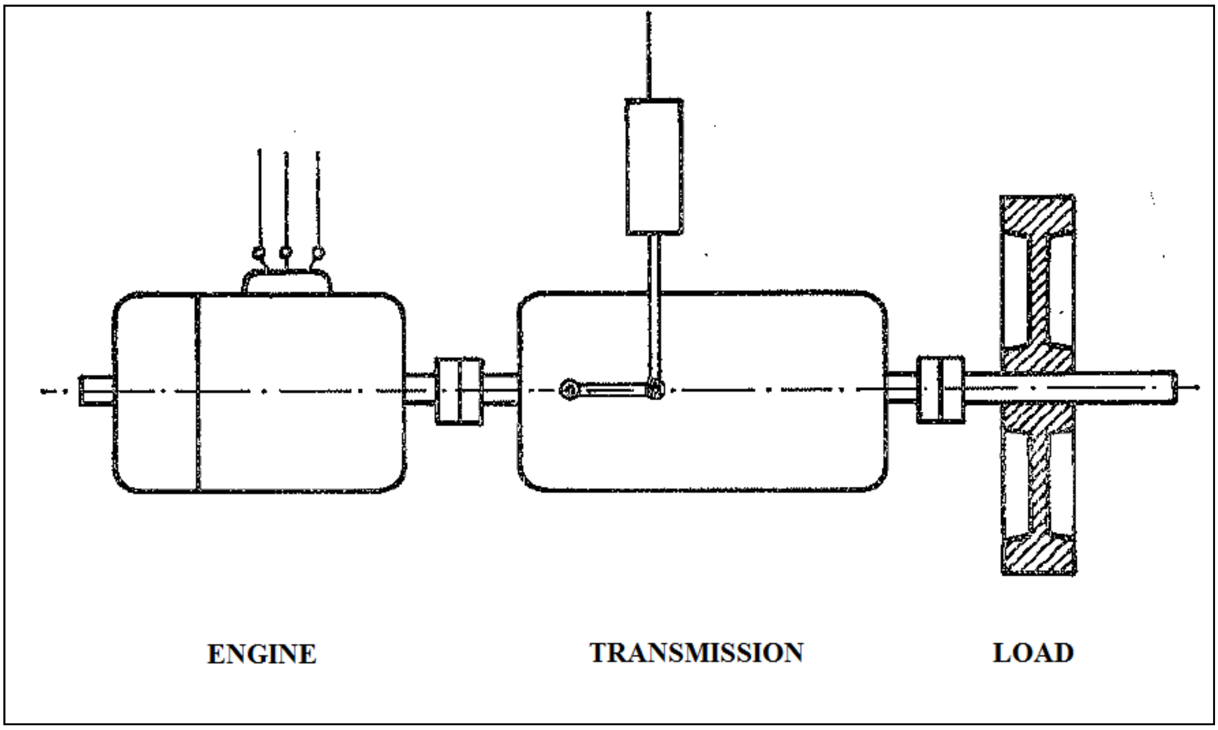

Figure 3-1: Diagram of the transmission system [1]

It is important to know that although speed is considered constant, this will be achieved only when an electric motor is used.

To understand how the system work, It will be used the diagram on Figure 3-2

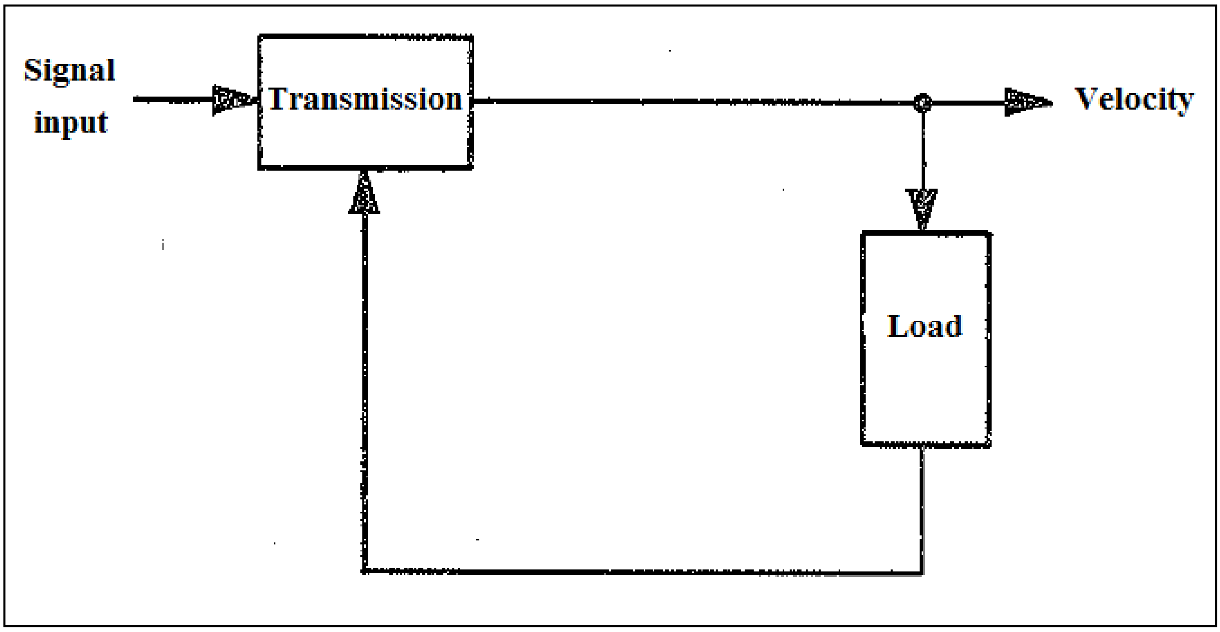

Figure 3-2: Block Diagram [1]

The position of the control lever (or control signal) will set the desired output speed. This output speed will produce a resistance torque, which will be function of the load. This torque also has an influence in the real output speed. That influence is quite little in hydrostatic transmission, however it is represented as a feedback circuit in the Figure 3-2.

Although the previous diagram is extremely useful, it is also possible to use another diagram, which is show in Figure 3-3 and is equivalent. In it, the transmission produces an output shaft torque which is a control signal function. As a result of that, the load achieves a rotary speed, which also influences in the real shaft torque. It is represented with the feedback circuit which is show in the next diagram.

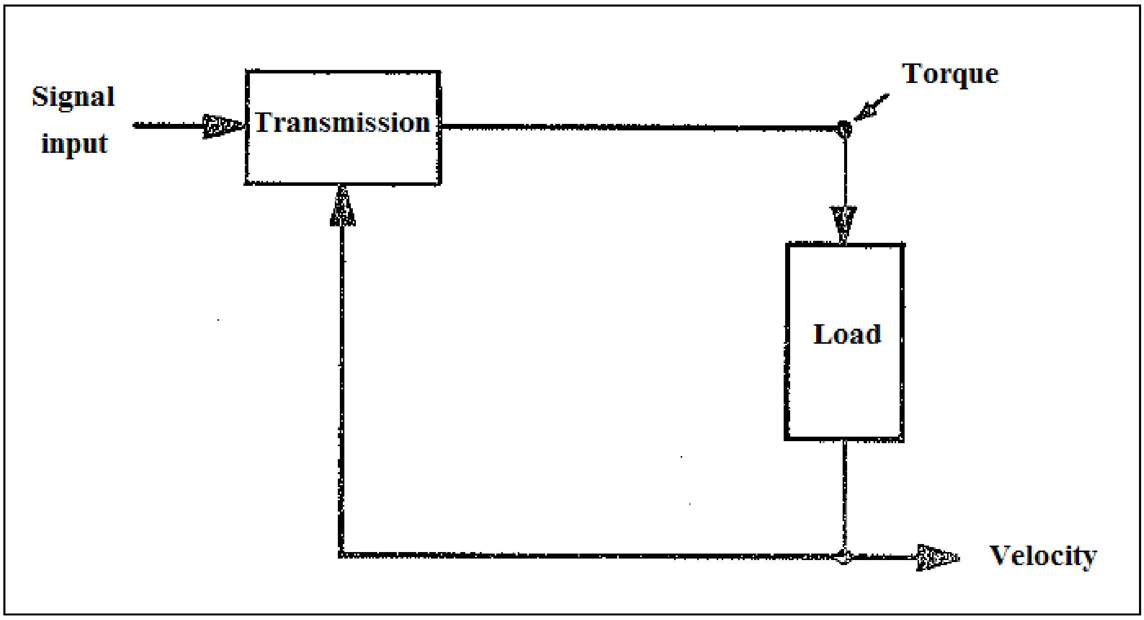

Figure 3-3: Equivalent block diagram [1]

In general, for hydrostatic transmissions, as the speed is almost independent of the load, are used the Figure 3-2.

3.2 Torque and apparent power

The next diagram shows the output torque depending on output speed for a variable velocity transmission, which has not losses and has a limited entry power.

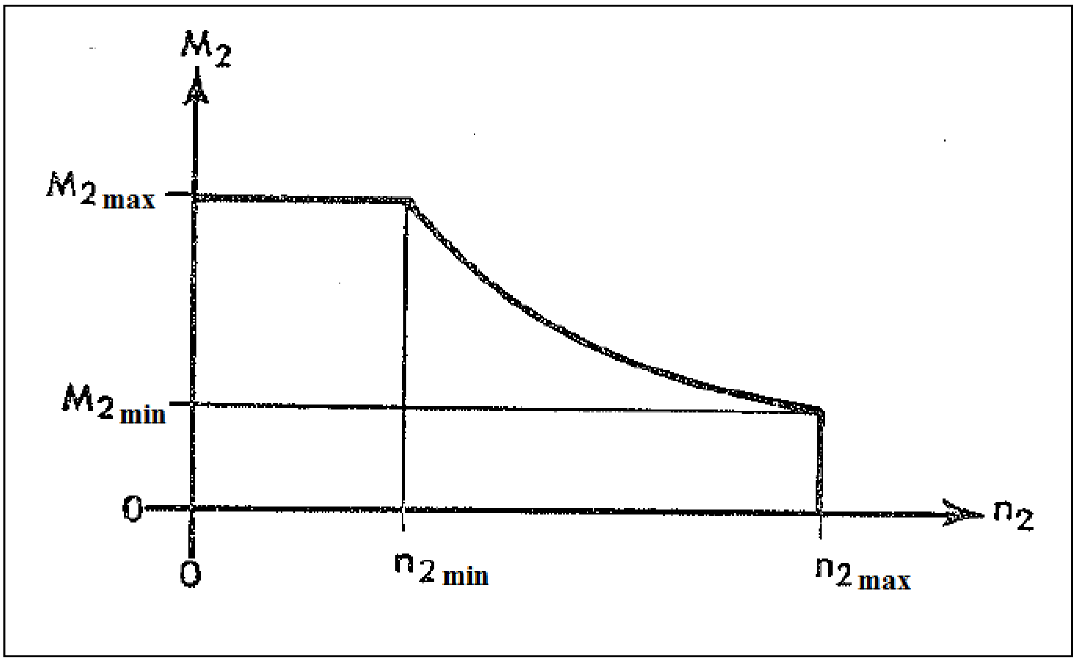

Figure 3-4: Torque depending on the output velocity [1]

Here it is possible to see how the maximum exit power is achieved with the lower output speed, where the power is enough. However, there is a velocity n_{2dir} where the output power starts to decrease because there is not enough entry power. Therefore, the lower output torque will be achieved with the maximum output speed. So if there are no losses, the output power will be the same as the input power.

$$P_{in} = M_{2max}\, n_{2min} = M_{2min}\, n_{2max} \tag{3-1}$$

It is common to use M_{2max} and n_{2max} as the main transmission magnitudes. So, it is possible to combine both in a magnitude called apparent power, which will not depend of a variation between the transmission and the load.

In the diagram below it is shown the available power depending on output speed, and thus the relation between shaft power and apparent power.

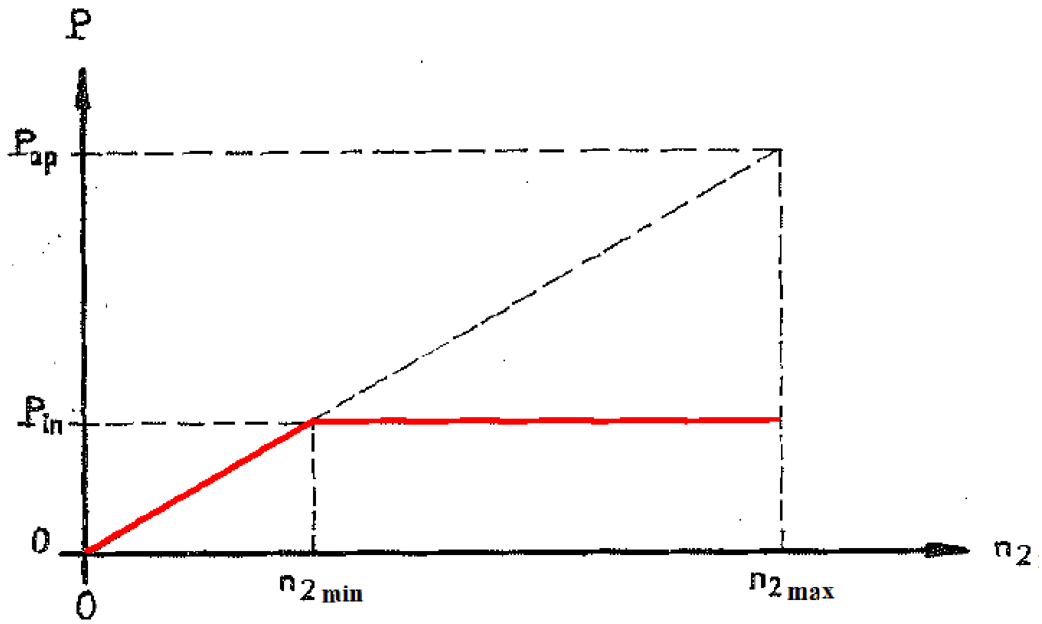

Figure 3-5: Power against output speed [1]

3.3 Gearbox

In the case of a gearbox with 3 gears, the diagram output torque against output speed will be the next.

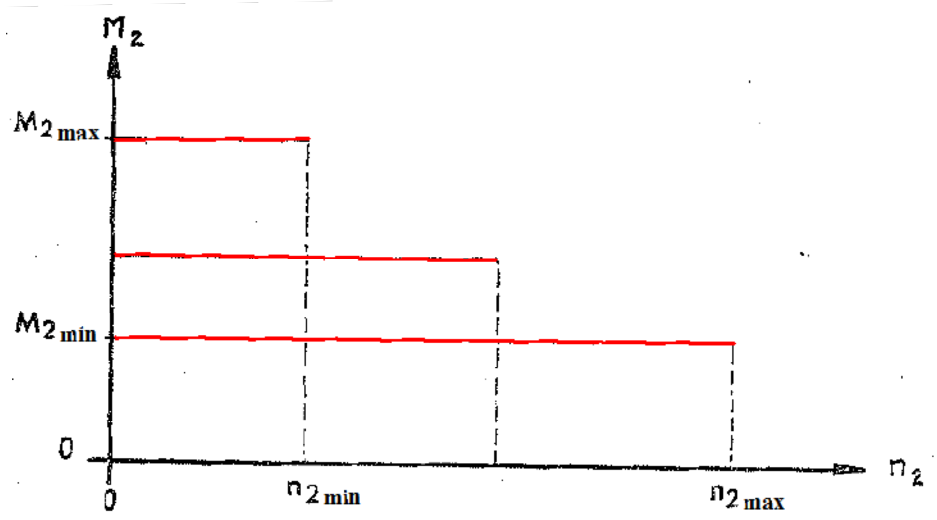

Figure 3-6: Torque depending on the output velocity for three gear ratios [1]

Here it is possible to see how the available torque grows when a lower gear is used for lower speeds.

In the next diagram, the available power depending on output speed is shown. Thus, all the power can be used in the output shaft for different output speeds.

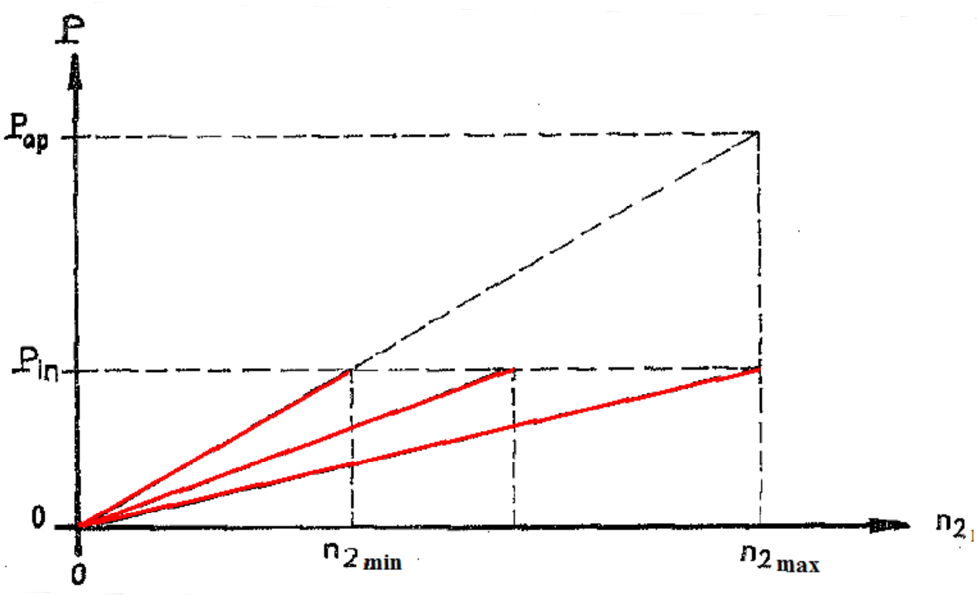

Figure 3-7: Power against output speed for three gear ratios [1]

At first, it is possible to use as many gears as needed. The only limitation will be the available place and the time needed to change gears.

Nevertheless, there is other really good option. That is, use a hydrostatic transmission, which have a huge output velocity range.

4. OBJECTIVE OF THE PROJECT

There are different objectives in this project that are going to be analysed, not only the study of the hydrostatic transmissions theory but the study of different aspects of the hydrostatic transmission simulating them with the simulation program AMESim and comparing with the expected theoretical results.

First of all the simulation is realized with the final objective of analyze it possible work for heavy machineries. They have the characteristic of a high requirement of productivity, output capacity and overall capacity. Also is important to know that their special design could need that the wheels axis has the possibility of changing a lot the position and then could be interesting a system which can send the engine power by flexible pipes instead of shafts.

A general view of the system that is going to be designed can be observed in the Figure 4-1 with a diesel engine which sends the power to the hydrostatic transmission. This hydrostatic transmission is made up of one variable pump and one variable motor. It sends a different shaft speed and torque depending on the displacement settings of both hydrostatic units to the machinery wheels but to have different possibilities it has a gearbox with at least two gear relations.

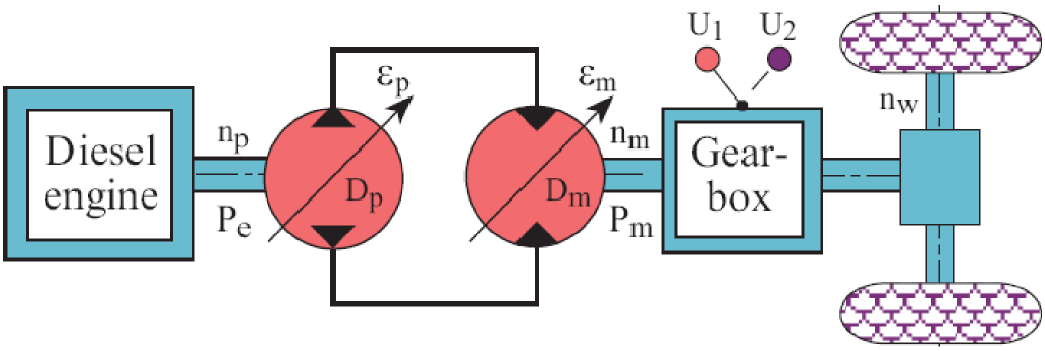

Figure 4-1: General schema of Hydrostatic Transmission for heavy vehicles [8]

For design the system showed on Figure 4-1 there has to be followed different steps.

The first thing that will be designed in the simulation program is the hydrostatic transmission, beginning on a basic system with only a pump a motor and a constant pressure system between the high and the low pressure pipes for then be sure that the transmission won't have a lower pressure that the one selected.

For selecting the hydrostatic units the parameters which are followed are the characteristics of the real transmission which is on the Linköping University Laboratory. Some of those data are for example the maximum displacement of the pump and the motor, 36 cm^3 and 56 cm^3 respectively, or the volume of the high pressure pipes and the low pressure pipes, which is roughly the same in both sides and equal to 550 cm^3, that is 0.5 litres.

Once the basic transmission is designed with it main characteristics will have to be designed systems to avoid the overloading of the system because of a high pressure, also a cooling system and an improvement of the low pressure system.

With a design of the hydrostatic transmission very close to the real one are going to be followed two different roads.

The first one is the design of a load as the one which is in the laboratory, with a system of valves making a Wheaston Bridge which can work in both directions for then analyse it results.

The other one is the design of a load close to the one that can have the heavy machinery, with the gearbox, high inertias and high torque corresponding to the wheels and the shafts of the vehicle.

Then will be analysed the resonance frequency and the dumping for different displacement settings of the motor and different gearbox transmissions comparing the simulation results with the expected theory data.

Finally the loop will be closed with different filters to allow the user the control of the final shaft speed instead of the control of the displacement setting of the pump.

5. SIMULATION MODEL OF THE HYDROSTATIC TRANSMISSION

In the following pages are going to be described the main stages of the simulation model. It is important to know that it begins from an extremely simplify model which only gives an easy idea of what is a hydrostatic transmission for then achieve step by step a final model really close to our test stand real hydrostatic transmission.

5.1 Stage 1

This is only a basic interpretation of the hydrostatic transmission with a hydrostatic variable pump and a hydrostatic variable motor.

The mechanical energy of the input shaft is transferred in a way of pressure energy of an incompressible liquid, and after is transformed again to mechanical energy in the output shaft.

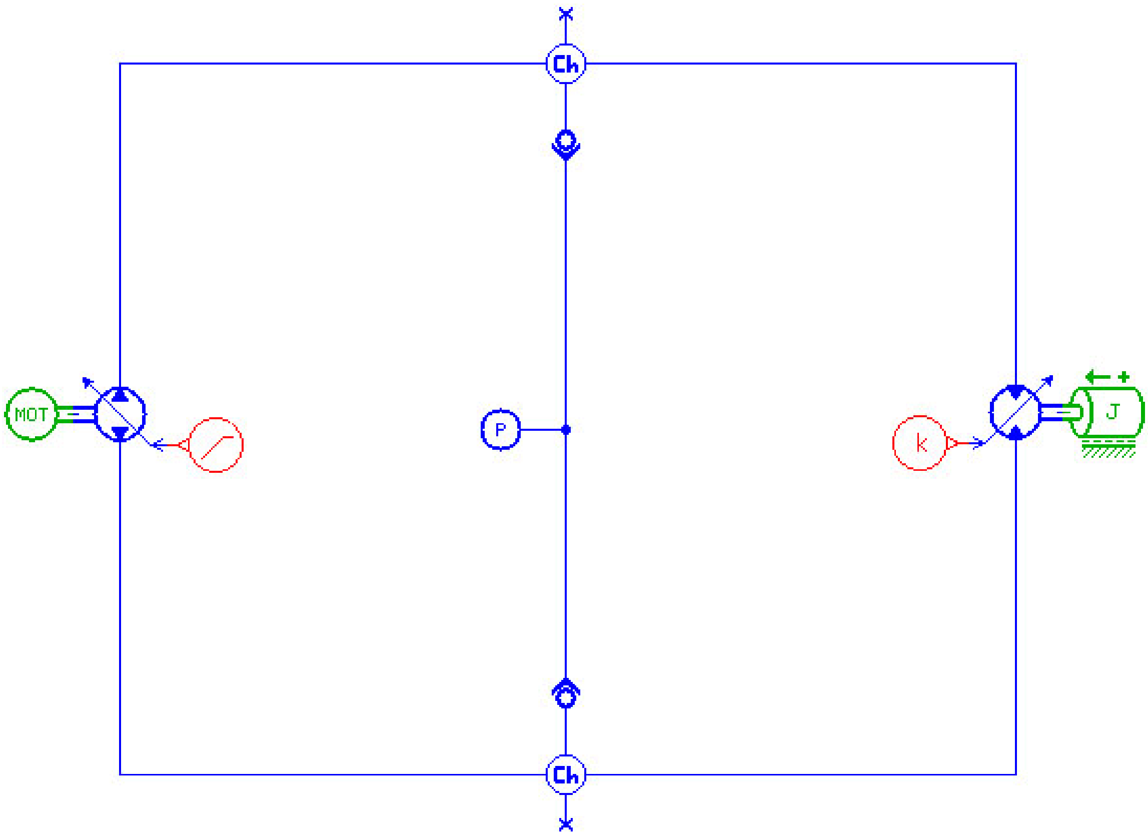

Figure 5-1: Basic hydrostatic transmission

It is important to see the constant pressure source which is connected with the high pressure pipe and the low pressure pipe to keep the desired pressure in both sides. This system also has two hydraulic check valves because it only sends the pressure from the constant pressure source to the main pipes. Finally there are two simple hydraulic chambers which are used to keep the same pressure on both sides of the connection.

To make a damping system which reduces the oscillations is decided to add a fixed rectangular hydraulic orifice in parallel to the hydrostatic pump and motor as the following picture.

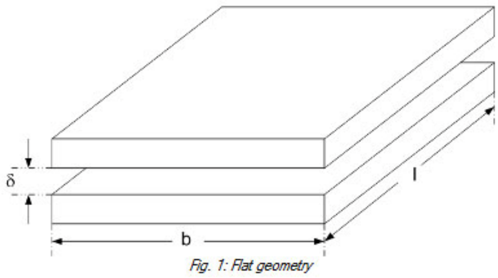

Figure 5-2: Fixed rectangular hydraulic orifice [19]

The other function of this system is to simulate the losses of all the circuit. There are two main kinds of losses. One of them is the loss due to the leakages of the system which represent the flow losses through the circuit. The other one is the torque losses due to dry and viscous friction and hydrodynamic effects.

In consequence the circuit for the first approximation after the introduction of all the elements that has been defined before, the fixed rectangular orifices for the pump and the motor, the inertia simulating the transmission inertias, and also the torque simulating the torque which simulate the torque losses can be observed on the following picture (Figure 5-3).

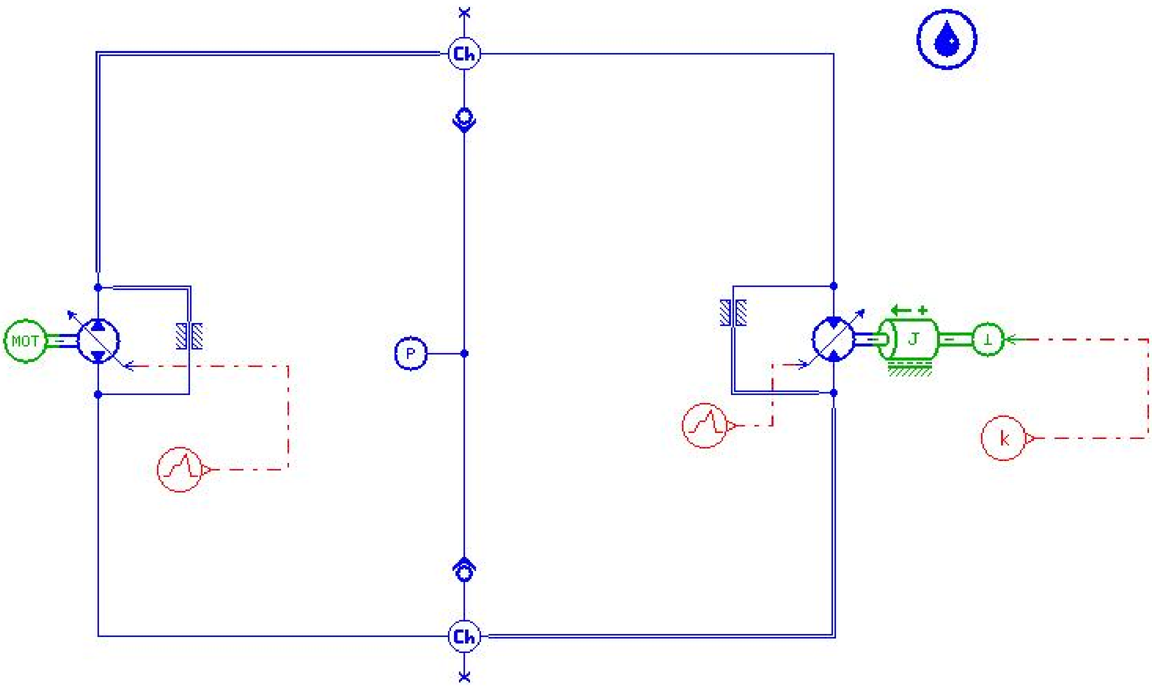

Figure 5-3: Hydrostatic transmission with flow and torque losses

The functions are introduced following the typical system for increase the flow in hydrostatic transmissions. This system consist on increasing the displacement setting of the pump linearly from 0 until it maximum value (1.0) and then decrease the displacement setting of the motor from its maximum value (1.0) until it minimum value (around 0.2) as in the following graphic where the red line represent the pump and the green line the motor.

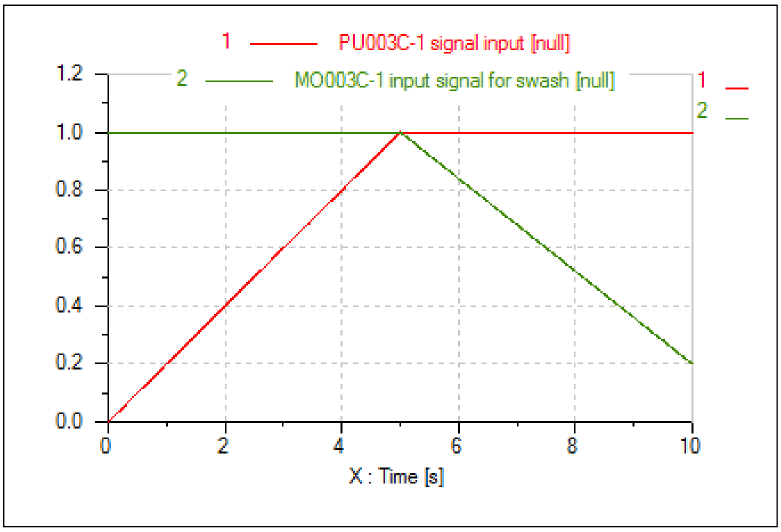

Figure 5-4: Displacement setting of the pump (red line) and the motor (green line)

The reason for this special distribution of displacement settings comes from the power definition for the pump and the motor and the relation between both of them.

$$P_p = \varepsilon_p D_p n_p \Delta p \tag{5-1}$$

$$P_m = \varepsilon_m D_m n_m \Delta p \tag{5-2}$$

An important detail is that the power in the pump and the motor is the same, so then the relation between equation (5-1) and equation (5-2) is the following one.

$$\frac{n_m}{n_p} = \frac{\varepsilon_p D_p}{\varepsilon_m D_m} \tag{5-3}$$

Where n_p and n_m are the speed of the pump and the motor, ε_p and ε_m the displacement settings of the pump and the motor, and D_p and D_m the displacement of the pump and the motor.

The displacement is constant in both hydrostatic units, so the only way to increase the speed relation is increasing the displacement setting of the pump and decreasing the same one in the motor. Is also important to know that the speed on the pump n_p is a constant, so then the objective is to increase the value of the speed in the motor.

The results obtained with the simulation in AMESim give some interesting information. The sign of the results will depend on the criteria selected before the simulation for each component but the important aspect is the absolute value.

All the results are with an input shaft speed of 1500 rev/min in the pump. Then the flow rate, which departure from the pump, follows the input signal and increase linearly until 55 litres per minute approximately (maximum value of the pump with this shaft speed) during the first 5 seconds. After, it will stay with no changes in this value until the end of the 10 seconds test (the red line at Figure 5-5). But the flow which goes to the motor (blue line) will decrease a little bit because of the flow which goes by the hydraulic orifice for damping the possible oscillations on the shaft speed and torque (green line).

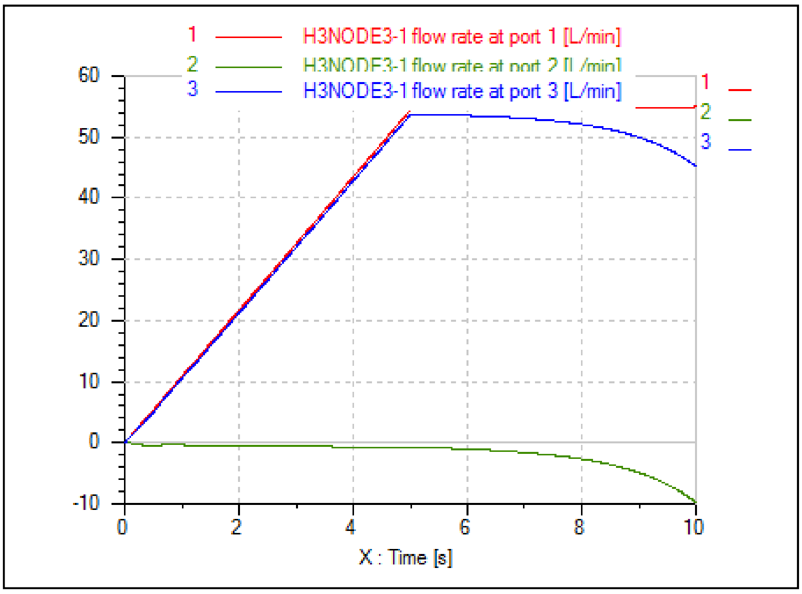

Figure 5-5: Flow rate at the hydraulic junction after the pump

Looking for the flow in the hydraulic junction before the pump is really important to see that the flow which is introduced to the pump (blue line in Figure 5-6) is again like in the input signal because the flow that went by the fixed hydraulic orifice (green line in Figure 5-6) is added to the one which comes from the motor (red line in Figure 5-6).

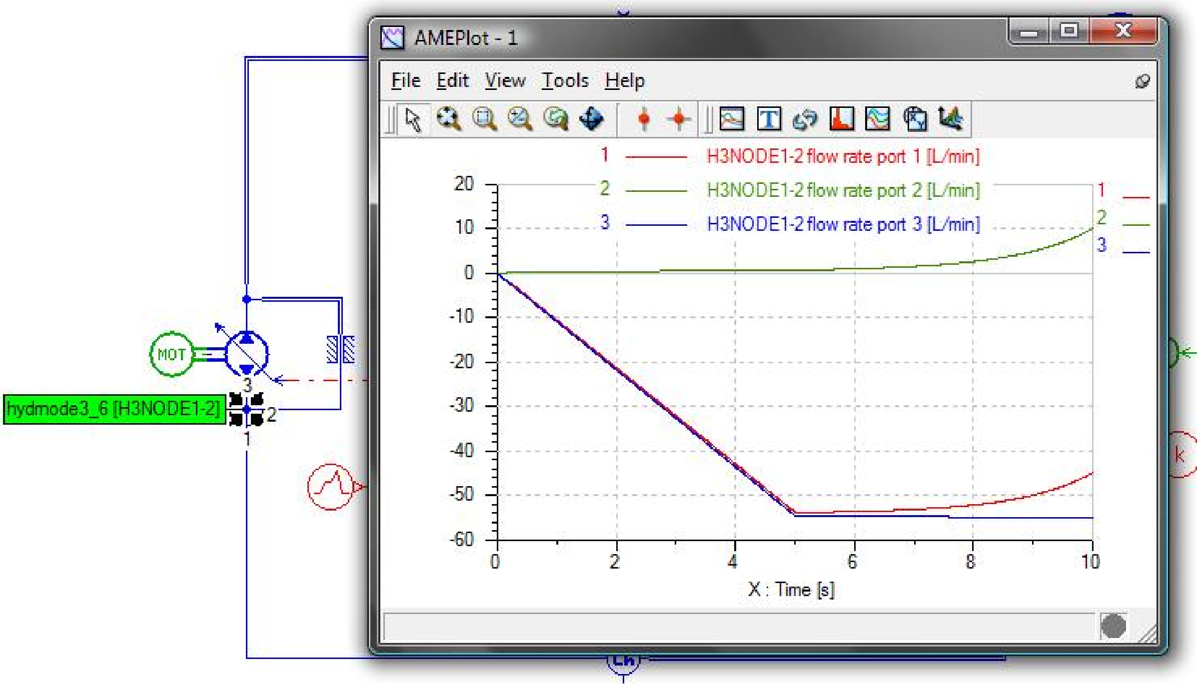

Figure 5-6: Flow rate at the hydraulic junction before the pump

The pressure of the liquid in the pump and the motor is different depending on the side of the circuit. In the high pressure side it grows like an exponential equation until the end of the simulation (10 seconds) (red line in Figure 5-7) while in the low pressure side it grows like the piecewise linear hydraulic pressure source (green line in Figure 5-7) which is between both sides, but that is because it is an extremely easy system which has to be improved in later stages.

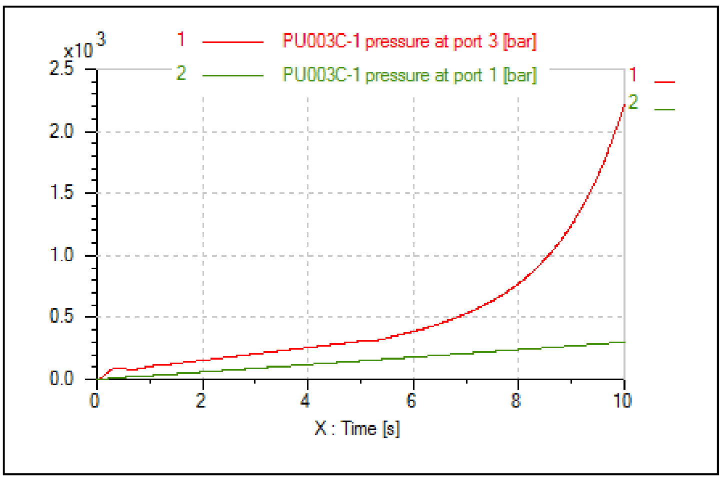

Figure 5-7: Pressure in the low pressure (green) and in the high pressure (red) sides

Finally, the shaft speed and the shaft torque can be compared between the model with the fixed hydraulic orifices and the model without them. The damping executed by the orifices makes that the shaft speed grows more linear (green line in Figure 5-8), which is the objective.

Looking for the shaft torque, the model with the orifices (green line in Figure 5-9) has a lower initial variation and also a more linear growing.

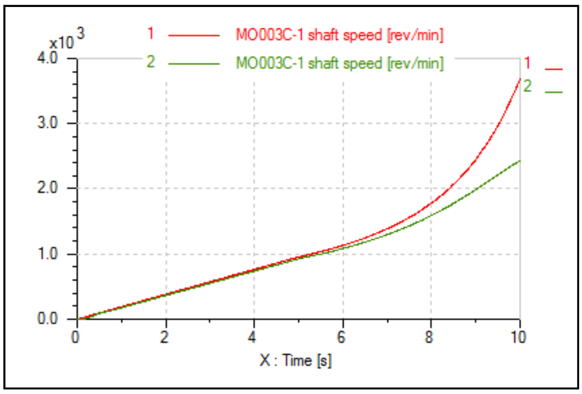

Figure 5-8: Shaft speed at the motor without orifices (red) and with them (green)

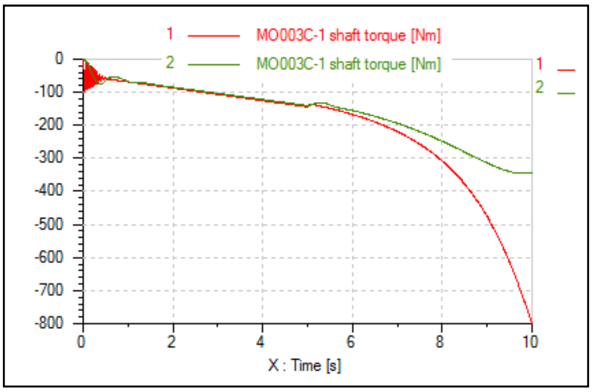

Figure 5-9: Shaft torque at the motor without orifices (red) and with them (green)

5.2 Stage 2

To make a better approximation to the real system it has been made a simulation as in the Figure 5-10 which has many new details that are going to be explained next.

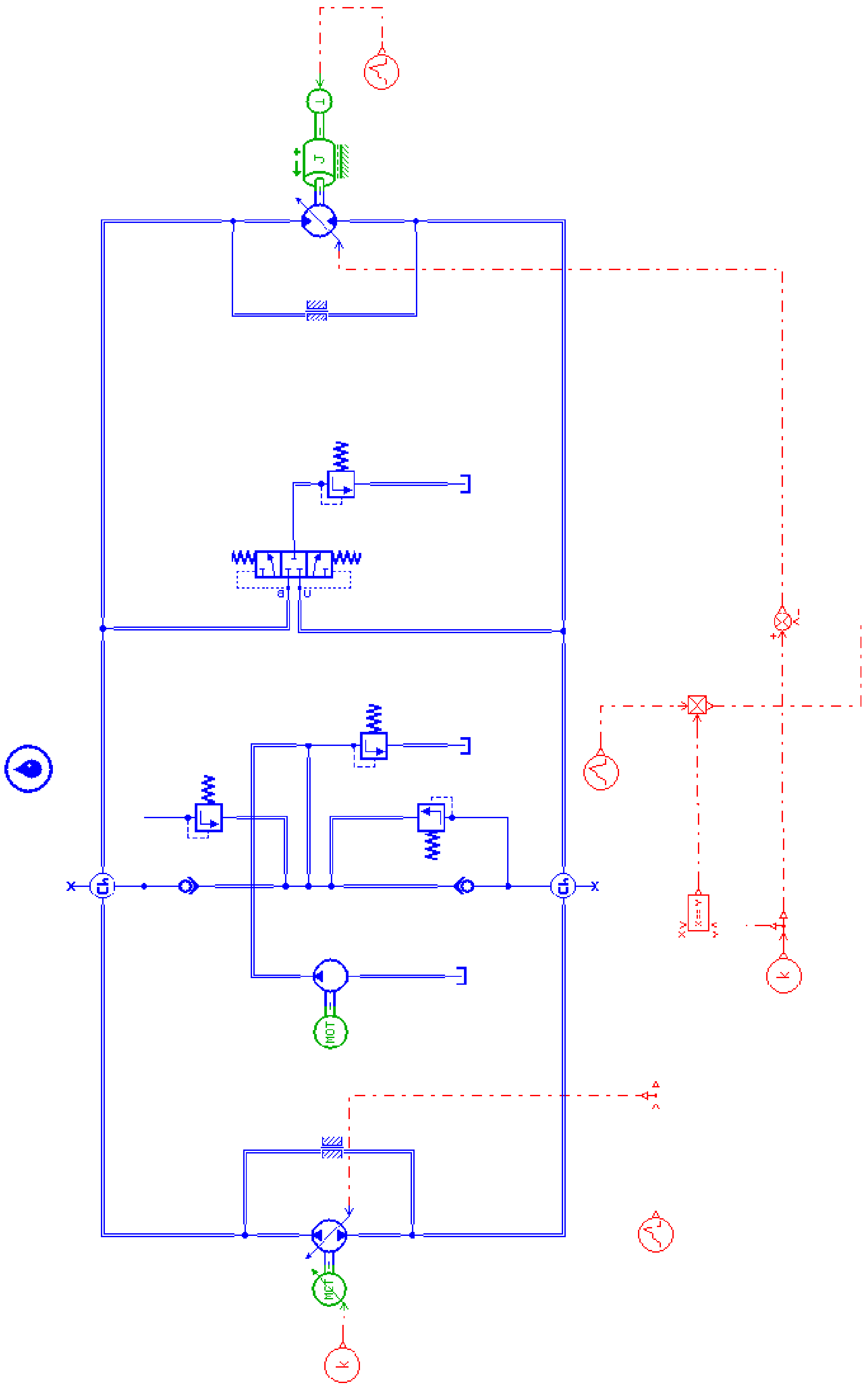

Figure 5-10: Laboratory real hydrostatic transmission simulation

The first element which has to be commented is the following one. It is formed by a booster pump (Figure 5-11 point 1), maximum pressure limitation valves (Figure 5-11 point 2) and anti-cavitation valves (Figure 5-11 point 3).

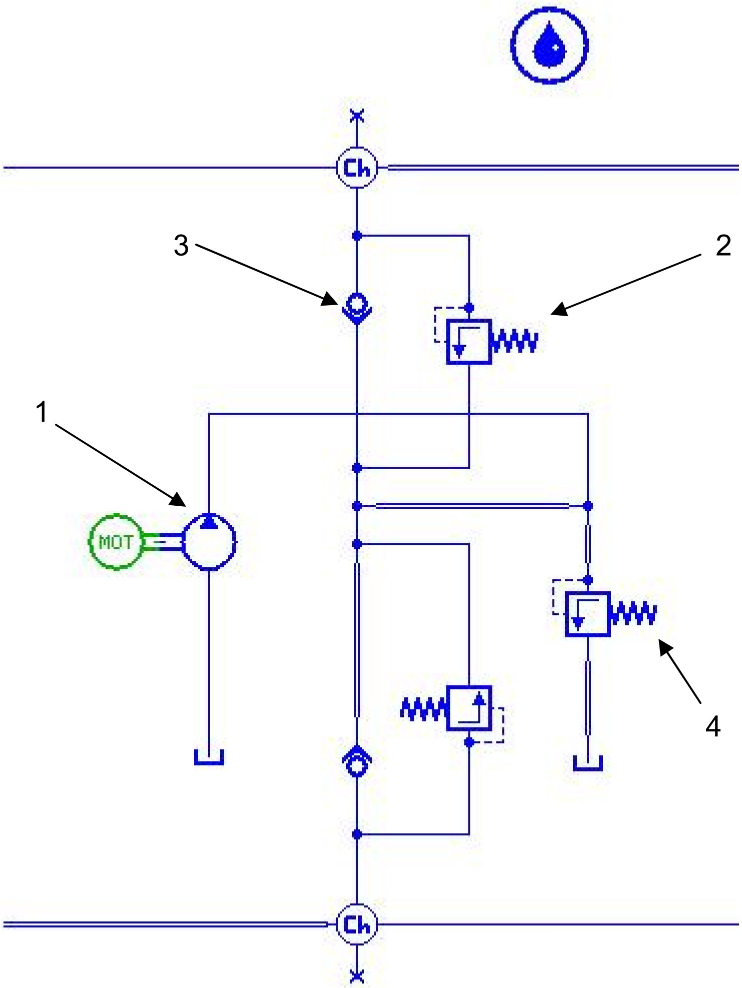

Figure 5-11: Maximum pressure limitation valves system

In first place is important to understand the mission of the booster pump, which is to set a constant pressure in the low pressure pipes of the hydrostatic transmission. This pump is connected to a constant speed prime mover, which has the same shaft speed than the one connected to the variable displacement hydraulic pump. It takes the hydraulic fluid from the hydraulic tank and then, in case that the pressure is lower than the relief valve cracking pressure (Figure 5-11 point 4), it goes until the low pressure side through the anti-cavitation valve. Otherwise the relief valve will open and part of the flow will return to the hydraulic tank. This route is represented in the Figure 5-12 by the red line.

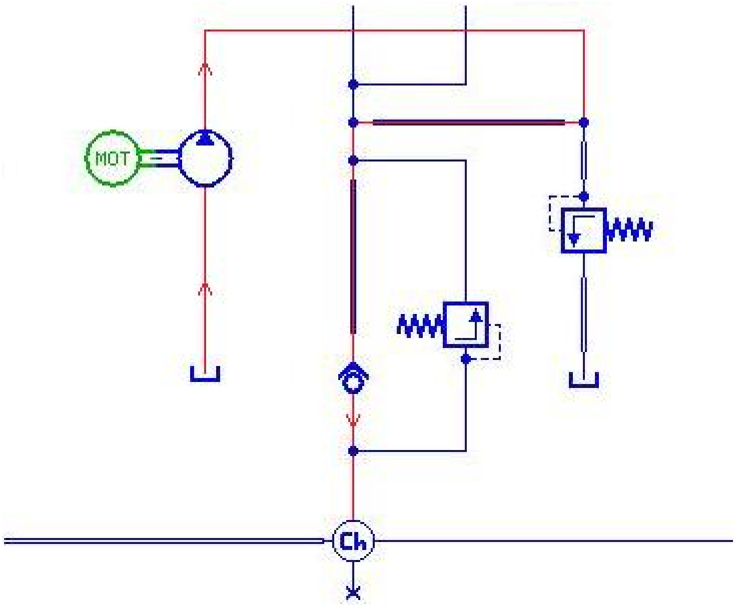

Figure 5-12: Constant pressure system for the low pressure pipes

The second function of the booster pump is to it as a cooling flow, which will be explained later on.

The main purpose of the Figure 5-11 is to limit the maximum pressure on the transmission high pressure side, and also if it is necessary on the low pressure side. To do it are used the pressure relief valves (Figure 5-11 point 2) and the anti-cavitation valves (Figure 5-11 point 3).

The pressure relief valves have a simple system which allow or avoid the flow rate depending on the pressure differences between both sides and the crack pressure.

The mode of the relief valve (open or closed) is determined by

$$dp = p_{in} - p_{out} - p_{crack} \tag{5-4}$$

If dp is positive, the relief valve is open and

$$q_{out} = dp \cdot grad \tag{5-5}$$

However, if it is negative, the valve is closed and

$$q_{out} = 0 \tag{5-6}$$

In either case,

$$q_{in} = -q_{out} \tag{5-7}$$

To understand better these equations it could be useful the following diagram (Figure 5-13)

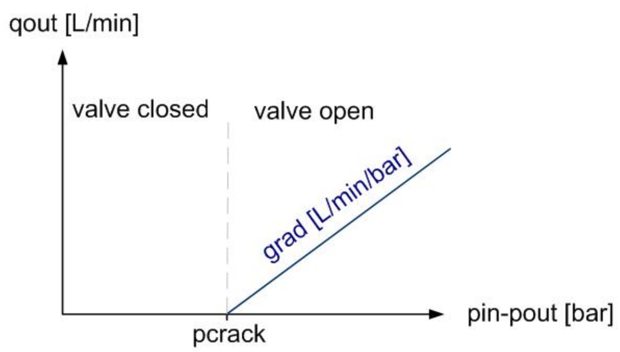

Figure 5-13: Operating system of a relief valve [19]

It is also added a cooling valve (Figure 5-14). It is controlled by the pressure difference which makes it open the low pressure side as in the Figure 5-15 where the higher pressure of side B opens the flow rate of side A.

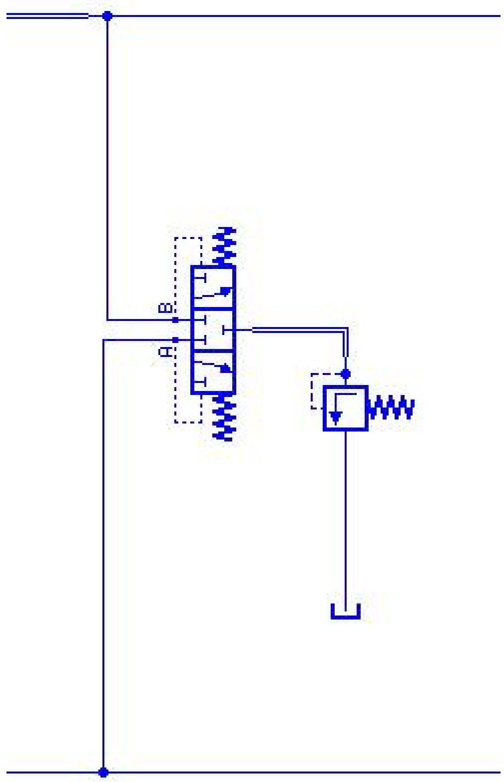

Figure 5-14: Cooling valve system

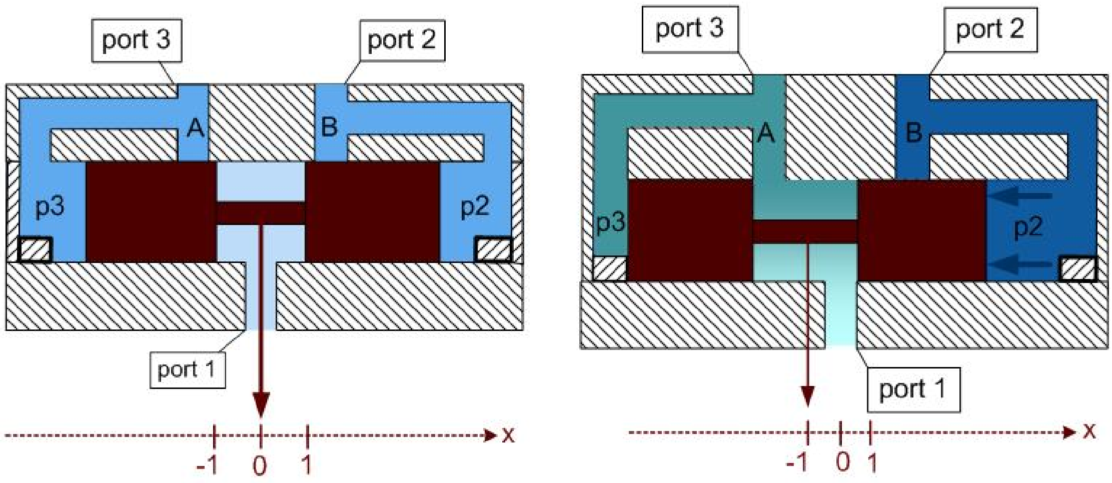

Figure 5-15: Working of the cooling valve [19]

The objective of the cooling system is to change part of hot hydraulic fluid which came from the motor side by new colder liquid which will appear from the booster pump. That is the reason why the booster pump also has a cooling function.

The crack pressure on the pressure relief valve of the cooling system must be lower than the one in the pressure limitation system (Figure 5-11 point 4). Otherwise the cooling system valve won't open and then the cooling system won't work.

In this case the hydrostatic transmission is still connected only to a low inertia and a constant torque value. Later it will be connected to a system which simulate the laboratory hydrostatic transmission load or to a gearbox with high inertia, depending of the purpose.

6. HYDROSTATIC TRANSMISSION WITH THE REAL LABORATORY LOAD

6.1 Analyse of the hydrostatic transmission

For simulate a real load is used a system like in the Figure 6-1. It is formed by different components.

The first one is a rotary load with two shafts which function is to simulate the inertia of the load connected to the output shaft. It has a small value because the test stand of this project has a low value.

Then it is connected to a fixed displacement bidirectional hydraulic pump which can send the hydraulic fluid to both directions depending on the rotary direction of the bidirectional hydraulic motor.

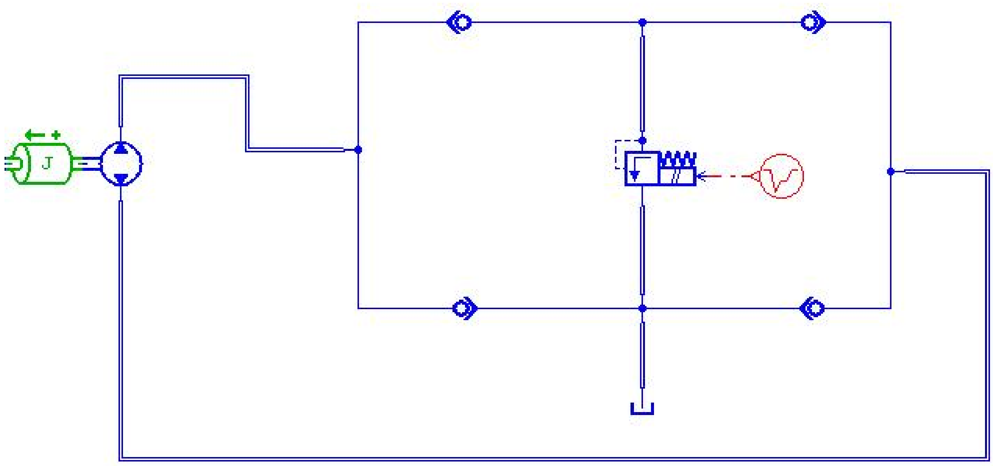

Figure 6-1: Simulation of the real system (Wheaston Bridge)

Finally there is a Wheaston Bridge which is formed by four hydraulic check valves and one pressure relief valve which has an input signal for varying the minimum pressure to open it and thus simulate the load. The check valves are connected to allow and avoid the both directions flow.

The simple signal operated pressure relief valve which is in the centre of the Wheaston Bridge does not open before a minimum pressure value for simulate the wheels torque.

This system has the special feature of a high damping value, something which makes the transmission very stable.

So then, it has to be connected to the output shaft of the motor directly to obtain a system as the one showed on the Figure 6-2.

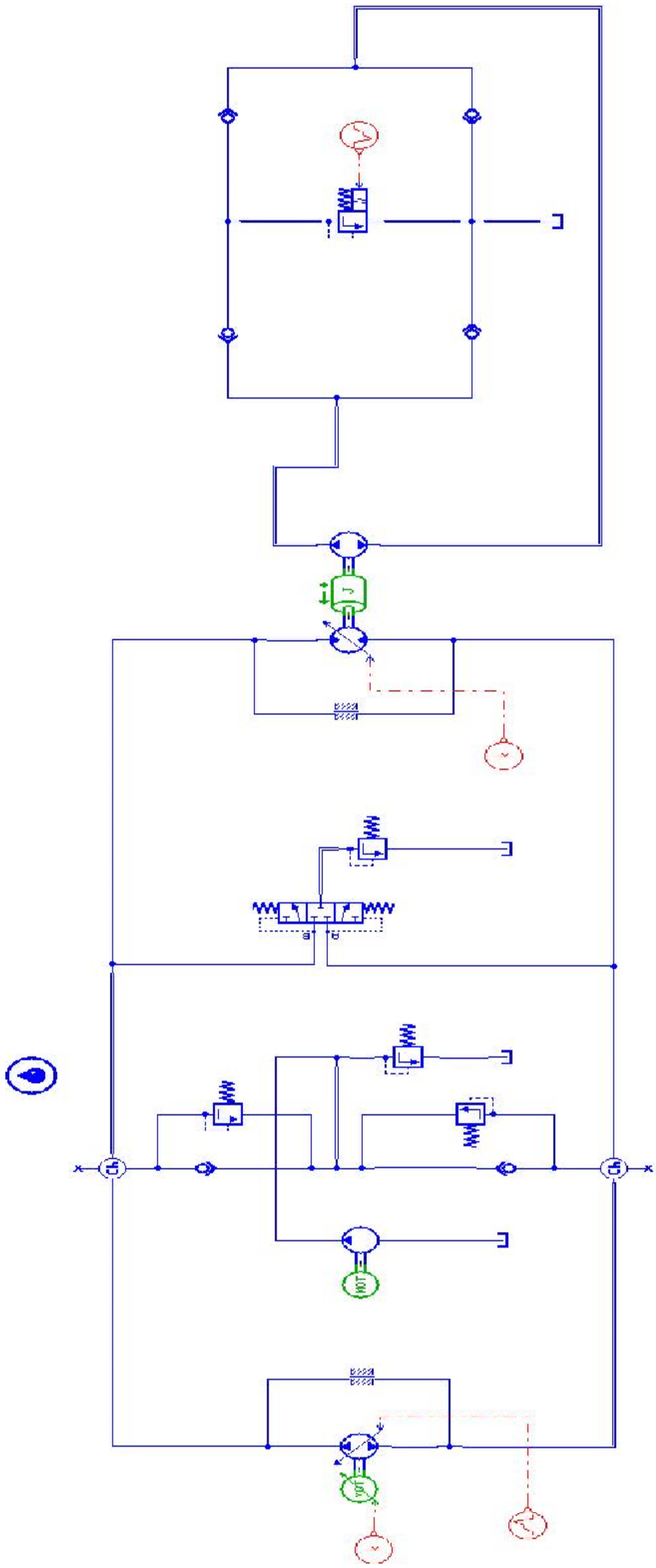

Figure 6-2: Simulation of Laboratory Hydrostatic Transmission

All this simulated transmission is characterized for the possibility of having two possibilities of working. The first possibility is as it has been explained before, where the fluid flows clockwise and then the first hydrostatic unit works as a pump and the second one as a motor. The other possibility is to work in the other direction where the first hydrostatic unit works as a motor and the second one as a pump. This can be used as a reverse gear or also as a braking system. Also the simulated output load could work in both directions due to the bidirectional hydraulic pump, and the special features of the Wheaston Bridge.

Moreover as could be seen on the previous diagrams, the whole system is symmetrical.

With this distribution the increasing of the displacement setting of the pump and the decreasing of the displacement setting of the motor like it was explained in the previous point with no problems of high oscillations because of the high damping of this load (Figure 6-3).

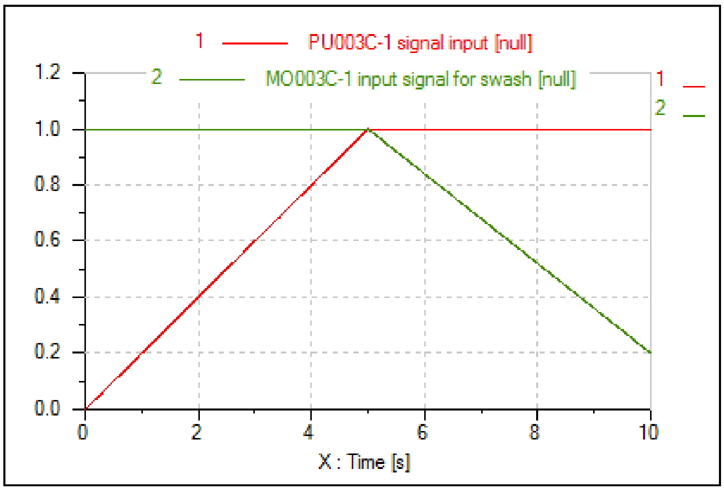

Figure 6-3: Displacement setting of pump (red) and motor (green)

Due to the displacement settings distribution the motor shaft speed will increase linearly with any important variation in the slope because of the damping (Figure 6-4). It is important to see that during close to 0.5 seconds the shaft speed is equal cero, which is because of the load characteristics. As has been explained before the load has a simple signal operated pressure relief valve which will be closed until a selected pressure, in this case 40 bar (figure 6-5), and then will allow the liquid circulation.

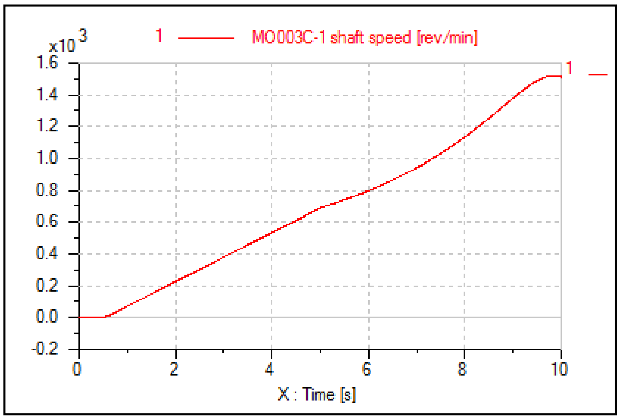

Figure 6-4: Output shaft speed

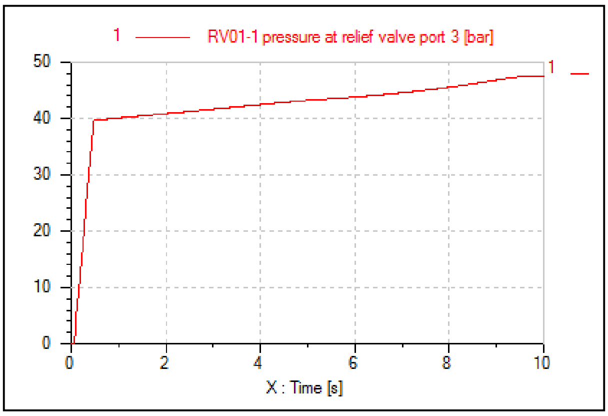

Figure 6-5: Relief valve pressure

It is also important to see that the pressure in the high pressure pipes will be stable from the moment that the simple signal operated pressure relief valve of the load is open until the moment that the displacement setting of the motor begin to decrease. From this moment the pressure will increase in an exponential curve until the safety high pressure system (p_{crack}), which opens the valve in the middle of the hydrostatic transmission (in this case at 322 bar). (Figure 6-6)

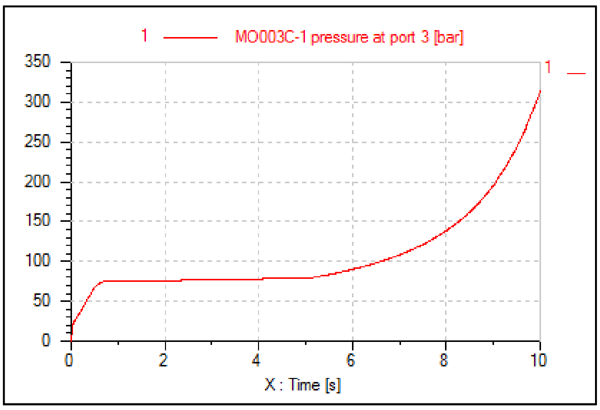

Figure 6-6: Pressure in high pressure side

The last important aspect to analyze is the value of the shaft torque. This value increases similar to the pressure in the load. That is because the objective of simulate the torque with the simple signal operated pressure relief valve. So then the torque will increase until a value around 40 Nm in 0.5 seconds for then continue growing slower (Figure 6-7) something similar to the relief valve pressure (Figure 6-5).

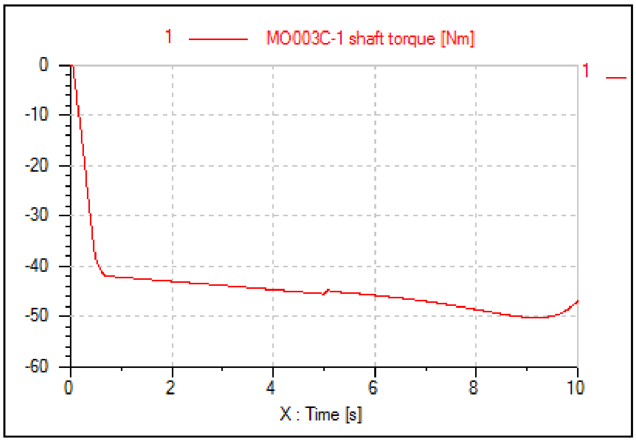

Figure 6-7: Output shaft torque

6.2 Reaction to a step input signal

Another important analysis which has to be done is the reaction of the hydrostatic transmission to a step in the displacement setting of the pump to check if is true the supposed high damping of this hydrostatic transmission configuration.

The idea is to increase the displacement setting of the pump until an intermediate value, in this case is going to be 0.4, wait there to be sure of the stabilization of the system for then increase as quick as possible to a higher value, in this case to 0.6 in 0.1 seconds (Figure 6-8). The displacement setting of the motor will be in one fixed value, in this case 1.0.

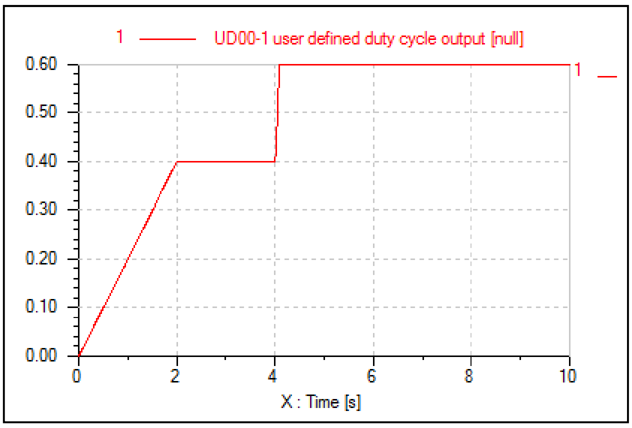

Figure 6-8: Displacement setting of the pump

Then it has to be analysed the reaction of the pressure in the motor high pressure side to see if it has a resonance frequency or otherwise the high damping avoid the oscillations.

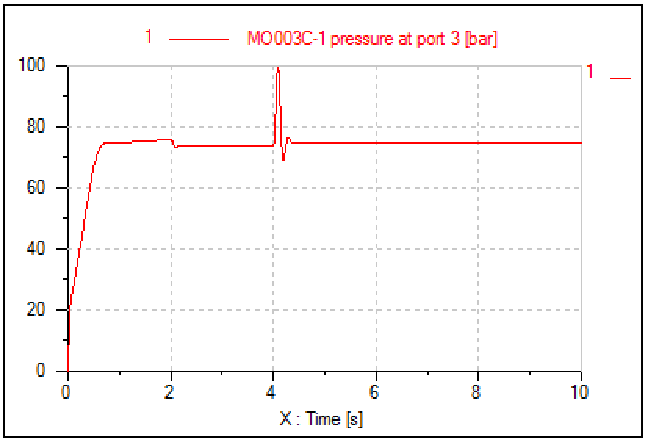

Figure 6-9: Motor pressure in the high pressure side

In the Figure 6-9 can be observed that the pressure in the second 4, when the displacement setting of the pump makes the step, has increased around 20 bar, but because of the high dumping it has never stayed oscillating and only performs one small insignificant oscillation which means that probably the value of the damping is bigger than 1.

7. CONNECTION TO A REAL VEHICLE

At this point, the main objective is to design a simulation of a heavy vehicle connected to the hydrostatic transmission from it output shaft.

There are many features that has to be taking into account for this design to allow the system the possibility of be like the real one.

First of all it is necessary to put an inertia which represents the different inertias of the hydrostatic transmission. It is the same that in the previous designs and it has to have a very low value because the hydrostatic transmission real parts also have low values (the global value is around 0.1kgm^2).

Once the inertia has been added it is necessary to connect it to a gearbox, which is one of the most difficult issues. The gearbox for this kind of vehicles is going to be designed with the possibility of engage two different gears, the first one to obtain the maximum traction force and the second one to obtain the maximum speed.

The program used for the simulation, AMESim has in it libraries some different gearbox designed (Figure 7-1) that can be added directly to the simulation.

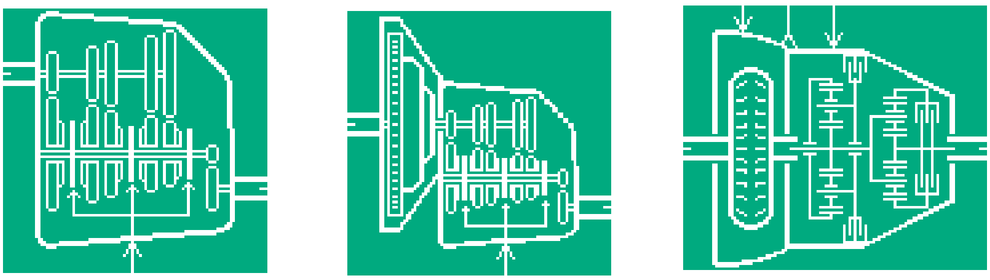

Figure 7-1: AMESim gearbox libraries [19]

The problem with those gearboxes, despite they can include the clutch and a lot of features, is that they are designed for more than four gear ratios, so then the simulations don't work as they are supposed to work.

The next option is to design a system with any element which can simulate the performance that the system should have with the gearbox. The best component to do it in the AMESim libraries is the modulated transformer rotary/rotary (Figure 7-2), but it is impossible to add with it the necessaries inertias, so the results are not completely successful using this component.

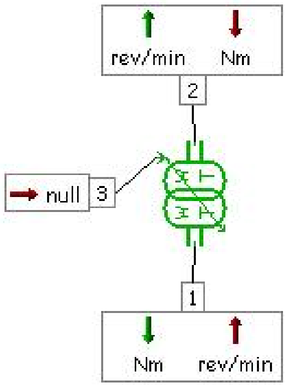

Figure 7-2: Modulated transformer rotary/rotary [19]

Finally the best applicable solution is to construct a completely gearbox. It is composed by two idle gears without inertia and losses, two synchronizers and two gear 3 ports. The result is showed in Figure 7-3. The reason for repeat the three elements twice is for the objective of has two different gear ratios.

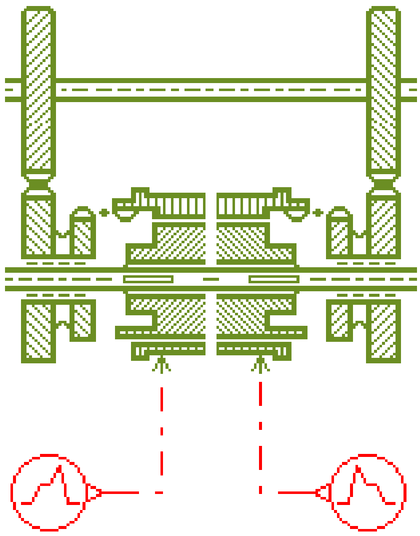

Figure 7-3: Simulation of a 2 gears Gearbox

This gearbox has the special feature of having all the pairs of wheels always in mesh. These kinds of gearbox are caller constant-mesh gearbox and a good example of them is showed in Figure 7-4. They have some advantages as the possibility of using “helical or double helical gear teeth which are quieter than straight teeth; it lends itself to the incorporation of synchronising devices more readily than the sliding-mesh box; the dog clutch teeth can be made so that they are easier to engage than the teeth of gear wheels, and any damage that results from faulty manipulation occurs to the dog clutch teeth and not to the teeth of the gear wheels.”

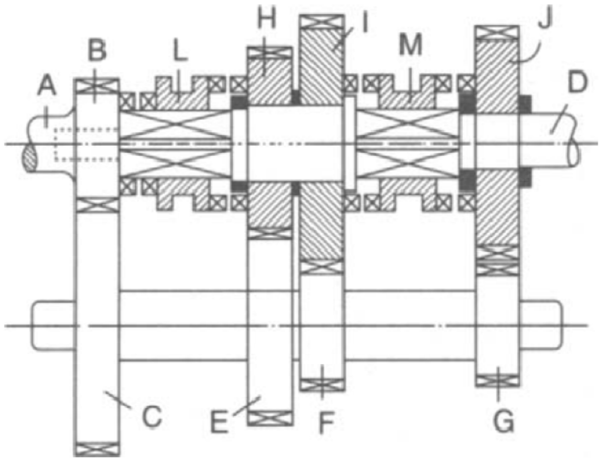

Figure 7-4: Constant-mesh gearbox [15]

Then has to be connected to the output gearbox shaft a new inertia corresponding to the wheels and all the inertias generated by the vehicle to the shaft. Due to those things it has a big value, around 100kgm^2. This inertia will change it action on the hydrostatic transmission depending on the gear selected. If the gear ratio is lower the effect of it will be bigger and then the damping will also increase.

Finally has to be added a torque source which represent the torque necessary to move the wheels. It is important to put it in opposition to the transmission torque because otherwise the simulation couldn't be real.

After all, the simulated gearbox with all it parts and the equivalent to the shafts and wheels are represented in the Figure 7-5 and also in the Figure 7-6 when it is added to the Hydrostatic transmission.

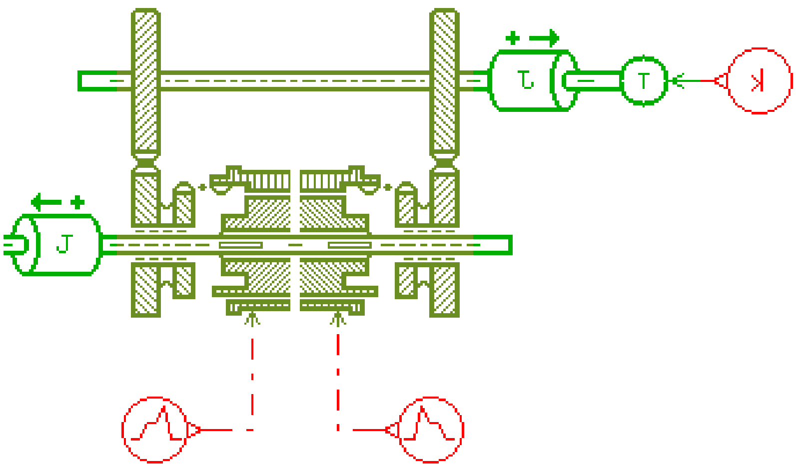

Figure 7-5: Connection to a real vehicle

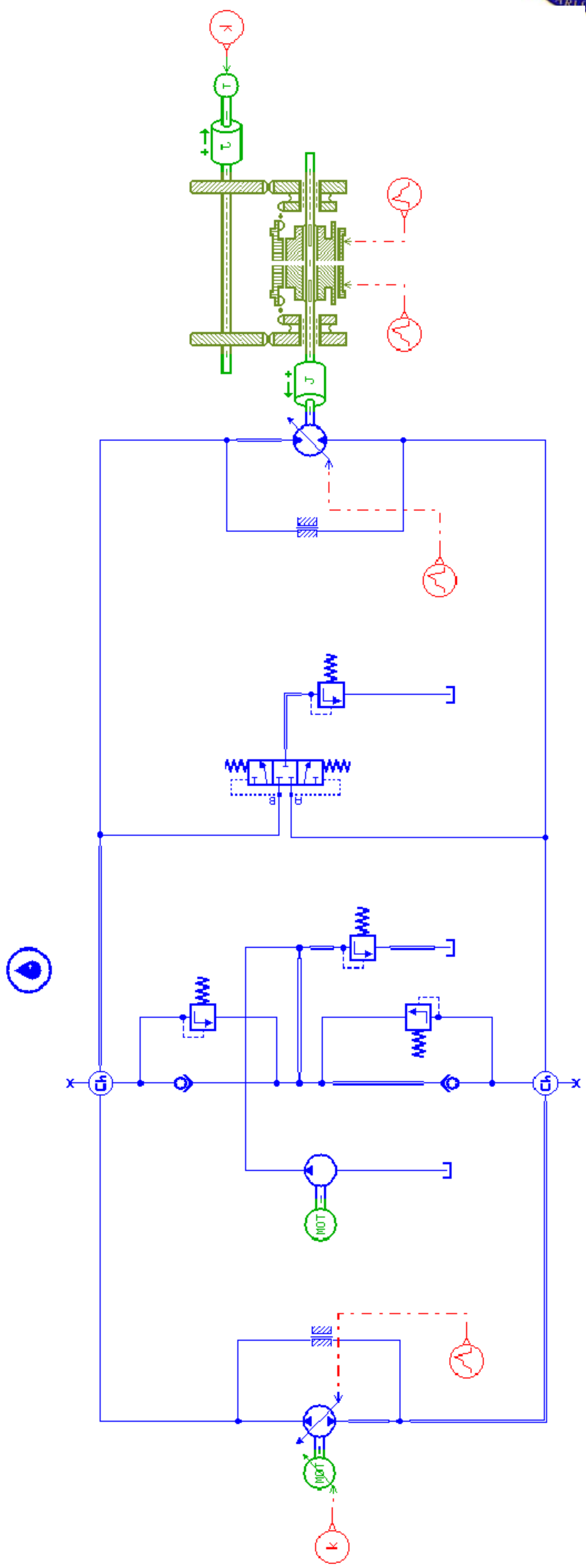

Figure 7-6: Hydrostatic Transmission connected to a real heavy vehicle

8. USE OF THE GEARBOX

The idea of the gearbox used for the hydrostatic transmission is to have the capacity of changing the gear ratio between the load and the transmission. Thus the final speed of the wheels shaft must be increased with no problems of overloading the transmission.

A general vision of the gearbox function can be observed in the Figure 8-1.

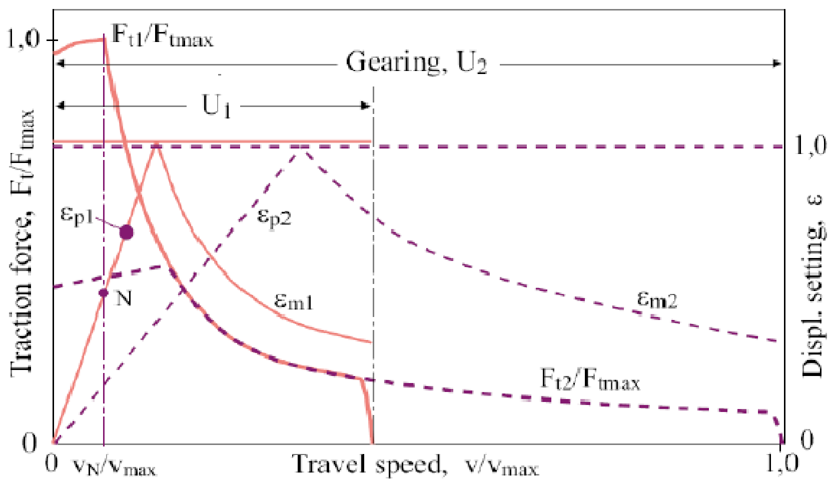

Figure 8-1: Force against velocity in a two gearbox hydrostatic transmission [13]

How is seen in the Figure 8-1, a gearbox in series with the hydrostatic transmission increase range and the power capacity without increasing the hydrostatic motor displacement.

Just as the gearbox has been designed, to engage the second gear ratio the synchronizers which are used as a clutch must disengage the idle gear of the first gear for, as soon as possible, engage the idle gear of the second gear.

In this case, the relations selected were $\frac{100}{10}$ for the first gear and $\frac{80}{12}$ for the second gearbox ratio.

The displacement setting of the pump and the motor are changed in different moments to avoid their coincidence with the gearbox changing and then obtain bad results from the simulation of difficult to recognise which one of the actions make the result change.

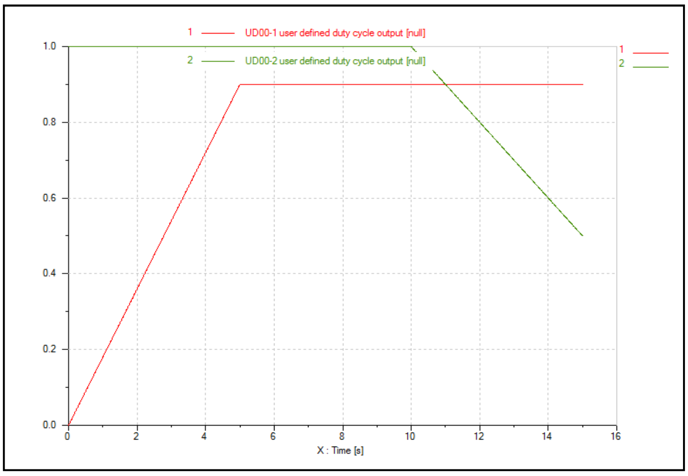

Figure 8-2: Displacement setting of the pump (red) and the motor (green)

Then the change of the gearbox ratio is decided to do at the second 7 to then don't have interferences from the pump or the motor.

In the first attempt the second gear is loaded by the clutch instantly as in the Figure 8-3 where the force applied by the synchroniser is the maximum possible. So with this system is supposed that the gearbox ratio is changed in cero seconds, something that is not real.

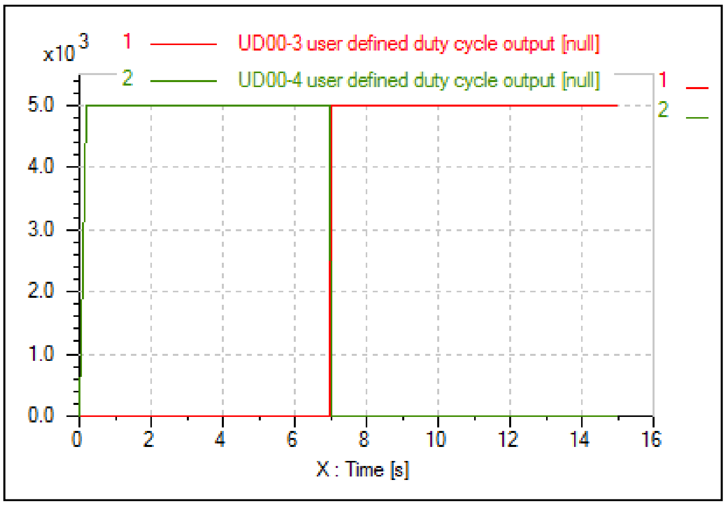

Figure 8-3: Force applied by the synchroniser to engage gear 1 (green) and 2 (red)

The results obtained from this simulation give some important information. The first important thing is that the shaft speed in the load side increase when the second gear is inserted and also when the output shaft speed of the hydrostatic transmission increase because of the displacement settings (Figure 8-4).

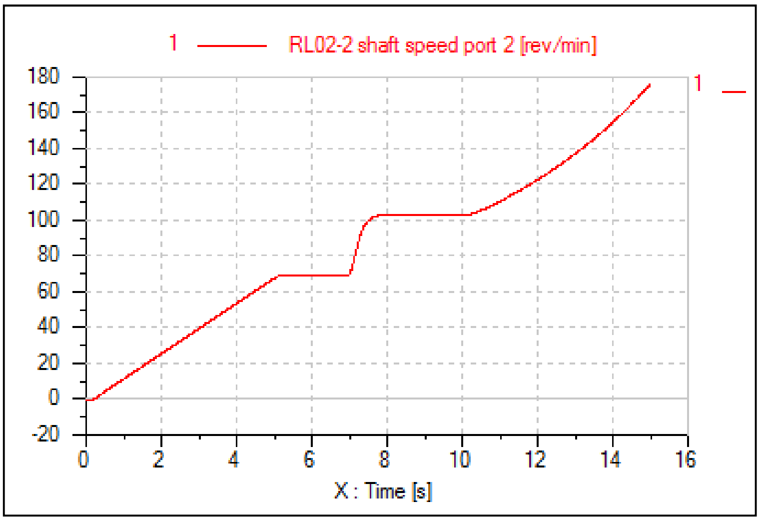

Figure 8-4: Shaft speed in the load side

The problem appears looking for the pressure and the torques applied in the system. The first one, the pressure in the high pressure side of the motor grows until high values close to uploading the hydrostatic transmission, and that is the reason why the torque in the load has a similar behaviour.

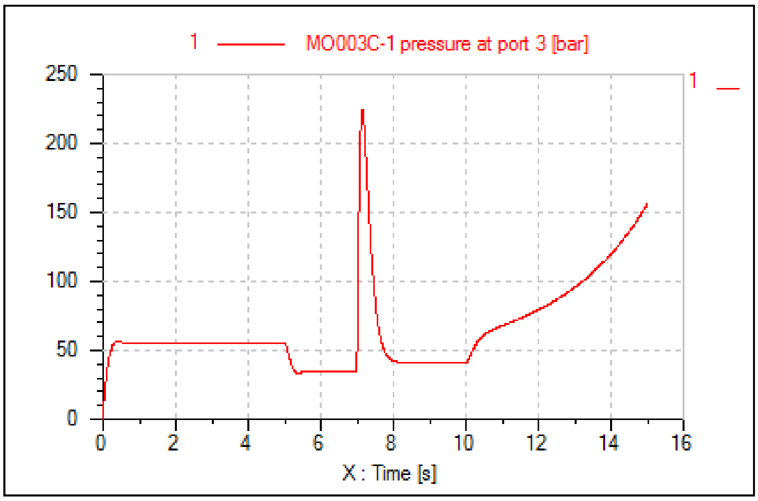

Figure 8-5: Pressure in high pressure side of the motor

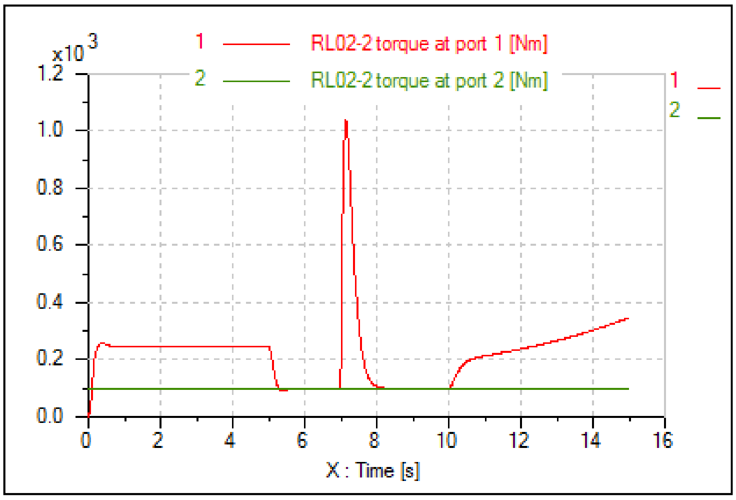

Figure 8-6: Torque in the secondary shaft of the gearbox before inertia (red) and after (green)

It can be observed that the torque which comes from the hydrostatic transmission has the same aspect than the pressure of the hydrostatic transmission. The increasing of its value is around ten times, so this is not acceptable to have great results.

To solve this problem the change of the gearbox ratio has to be done gradually until the maximum value. The method for doing it is to make the force with the second gear synchroniser also gradually (Figure 8-7).

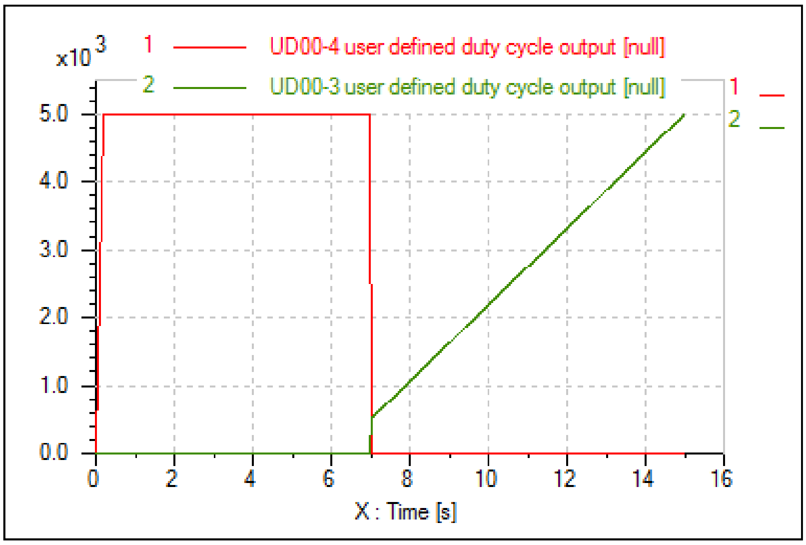

Figure 8-7: Force applied by the synchroniser to engage gear 1 (red) and 2 (green)

So then the result obtained by the engage of the gears as in the Figure 8-7 has improved and they can be considered as in a real system. To compare with the previous gearbox ratio changing the figures 8-8 and 8-9 have a value in the moment of the gearbox changing of only around two times bigger, a reasonable one.

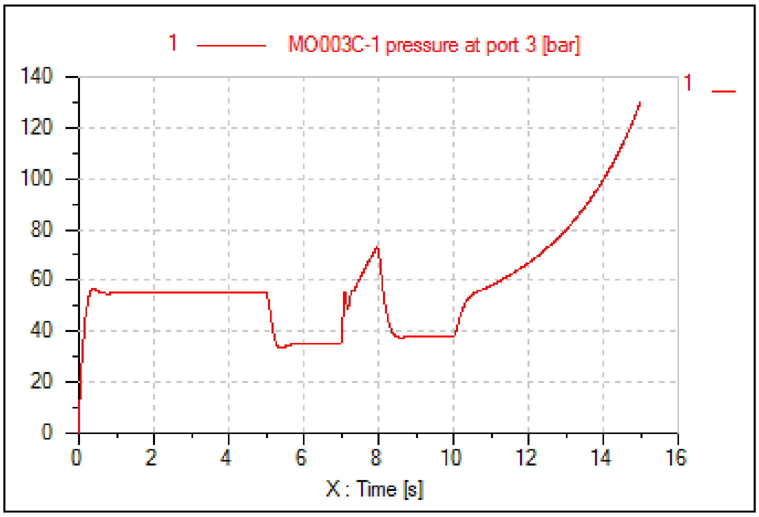

Figure 8-8: Pressure in high pressure side of the motor

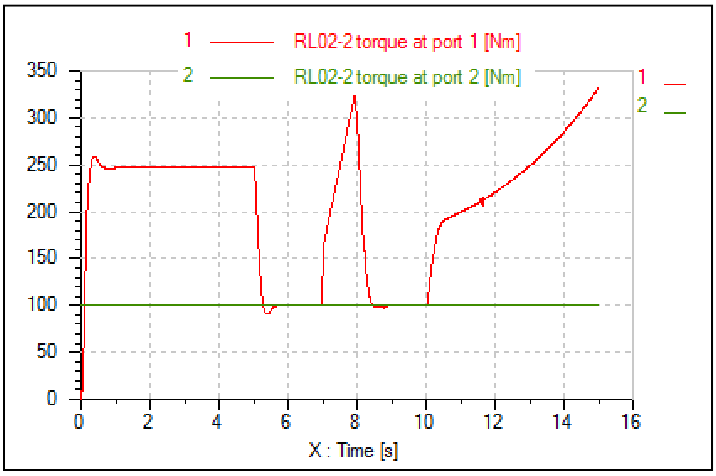

Figure 8-9: Torque in the secondary shaft of the gearbox before inertia (red) and after (green)

9. DYNAMIC ANALYSES AND CONTROL OF THE HYDROSTATIC TRANSMISSION

9.1 Dynamic equations

During normal operation the pressure in one line between pump and motor will be at replenishing pressure (p_r=p_2) and the other pressure will modulate to match the load (p_1) (supposing the high pressure pipes in line 1). The two lines will switch functions if the load dictates a pressure reversal. It is possible for both line pressures to vary simultaneously if transients are rapid and load reversals occur. However, for system modelling it is assumed that only one pressure varies, the high pressure line, at the same time and both sides are identical.

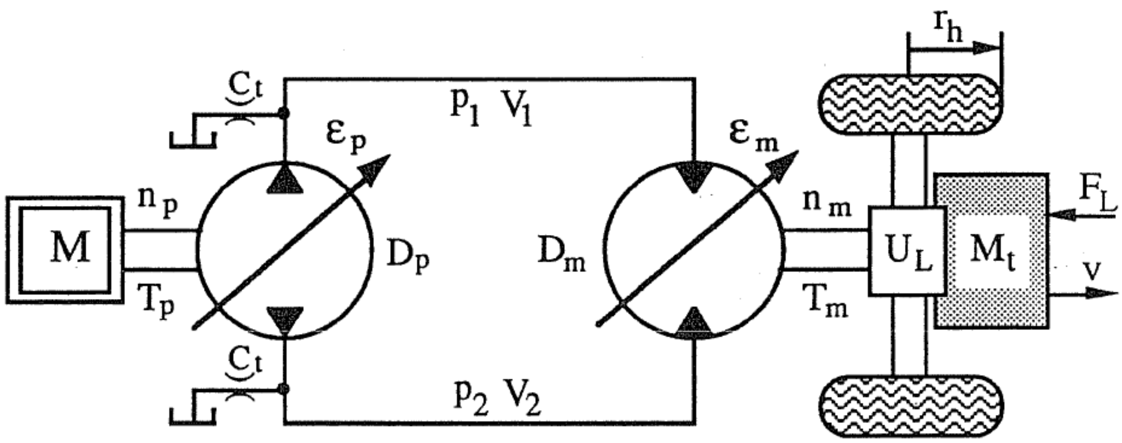

Figure 9-1: Hydrostatic transmission with gearbox applied on a vehicle [13]

Applying the continuity equation for the volume V_1:

$$\Delta\varepsilon_p D_p \omega_p - C_{tp}\Delta P_1 - C_{tm}\Delta P_1 - \Delta\varepsilon_m s D_m \Delta\theta_m = \frac{V_1}{\beta_e} s\Delta P_1 \tag{9-1}$$

Introducing the total leakage coefficient for pump and motor, $C_t = C_{tp} + C_{tm}$ gives

$$\Delta\varepsilon_p D_p \omega_p - C_t\Delta P_1 - \Delta\varepsilon_m s D_m \Delta\theta_m = \frac{V_1}{\beta_e} s\Delta P_1 \tag{9-2}$$

Now it is going to be analysed the torque balance equation for the motor shaft. To do that, it is going to be used the Newton second law. If, the friction torque is described only by a viscous friction coefficient, B_m the torque equation is given as

$$D_m(\Delta P_1 - \Delta P_r) = [P_r = constant] = D_m \Delta P_1 = J_t s^2 \Delta\theta_m + B_m s \Delta\theta_m + \Delta T_L \qquad (9\text{-}3)$$

From these equations and also with the analysis of the feedback (which is going to be explained later) it is possible to obtain the main equations for the resonance frequency ω_h and the damping δ_h (9-4 and 9-5).

$$\omega_h = \sqrt{\frac{\beta_e \varepsilon_m^2 D_m^2}{V_1 J_t}} \qquad (9\text{-}4)$$

$$\delta_h = \frac{C_t}{2\varepsilon_m D_m}\sqrt{\frac{\beta_e J_t}{V_1}} + \frac{B_m}{2\varepsilon_m D_m}\sqrt{\frac{V_1}{\beta_e J_t}} \qquad (9\text{-}5)$$

J_{total} will be calculated as it appears in the equation (9-6)

$$J_t = J_m + \frac{J_L}{U_m^2} \qquad (9\text{-}6)$$

Where J_m is the inertia of the motor and hydrostatic transmission, J_L is the inertia of the load and U_m is the gearbox ratio which is calculated as in the equation (9-7).

$$U_m = \frac{n_m}{n_L} > 1{,}0 \qquad (9\text{-}7)$$

As is shown in the equation (9-5) the damping is direct proportional to the transmission leakage coefficient, C_t

$$\delta_h \sim C_t \qquad (9\text{-}8)$$

To calculate the value of the transmission leakage coefficient it is necessary the calculation of the leakage flow. This value will be obtained from the volumetric leakage flow from the orifices and from the efficiency of the motor and the pump.

$$q_{ep} = \varepsilon_p D_p n_p \left(1 - \frac{q_{lep}}{\varepsilon_p D_p n_p}\right) \qquad (9\text{-}9)$$

$$\eta_{vp} = \left(1 - \frac{q_{lep}}{\varepsilon_p D_p n_p}\right) \qquad (9\text{-}10)$$

$$q_{lep} = (1 - \eta_{vp})\varepsilon_p D_p n_p \tag{9-11}$$

Using the same procedure can be obtained the value of the efficiency leakage flow of the motor q_{lem}

$$\eta_{vm} = \left(\frac{1}{1+\frac{q_{lem}}{\varepsilon_m D_m n_m}}\right) \tag{9-12}$$

$$q_{lem} = \left(\frac{1}{\eta_{vm}} - 1\right)\varepsilon_m D_m n_m \tag{9-13}$$

Finally, to obtain the value the transmission leakage coefficient it is used the equation (9-14)

$$C_t \Delta P = q_{ltot} = q_{lep} + q_{lop} + q_{lem} + q_{lom} \tag{9-14}$$

Nevertheless the value of the transmission leakage coefficient in this case is considered as a constant value because of linearised model.

9.2 Study of resonance frequency and damping

Once the simulation was correct it is the moment to obtain the information from it. To do that, it is going to be varied the gear ratio and the displacement setting of the motor, and thus get many different working point of this transmission.

To achieve the correct simulation, it has to be set some different parameters. These are taken from the Linköping Laboratory's hydrostatic transmission and are showed on the Table 9-1, Table 9-2 and Table 9-3.

Pump

Efficiency	μ_p	0,99	[-]		
Leakage Flow (efficiency)	q_{lep}		l/min	*1/(6*10^4)	m^3/s
Leakage Flow (orifice)	q_{lop}		l/min	*1/(6*10^4)	m^3/s
Leakage Flow (pump total)	q_{ltp}		l/min	*1/(6*10^4)	m^3/s
Volumetric Displacement	D_p	36	cc/rev	5,72958E-06	m^3/rad
Displacement setting	ε_p		[-]		
Rotary Velocity	n_p	1500	rev/min	157,0796327	rad/s
Total Leakage Coefficient	C_{tp}		m^3/Pas		
Viscous friction coefficient	B_p	0	Ns/m		

Table 9-1: Parameters of the pump

Motor

Efficiency	μ_m	0,99	[-]		
Leakage Flow (efficiency)	q_{lem}		l/min	*1/(6*10^4)	m^3/s
Leakage Flow (orifice)	q_{lom}		l/min	*1/(6*10^4)	m^3/s
Leakage Flow (pump total)	q_{ltm}		l/min	*1/(6*10^4)	m^3/s
Volumetric Displacement	D_m	56	cc/rev	8,91268E-06	m^3/rad
Displacement setting	ε_m		[-]		
Rotary Velocity	n_m		rev/min	*2*pi()/60	rad/s
Total Leakage Coefficient	C_{tm}		m^3/Pas		
Viscous friction coefficient	B_m	0	Nms/rad		

Table 9-2: Parameters of the motor

Hydrostatic Transmission

Pressure increment	ΔP		Pa		
Effective Bulk Modulus	β_e	10000	Bar	1E9	Pa
Volume	V_1	550	cc	5,5E-4	m^3
Capacitances in transmission lines V_1/β_e	C_1	5,5E-13	m^3/Pa		
Inertia	J_t		kgm^2		
Resonance Frequency	ω_h		rad/s		
Hydraulic Damping	δ_h		[-]		
Gearbox ratio	U_m		[-]		
Total Leakage Coefficient	C_t		m^3/Pas		
Highest process speed Gear Ratio	U_{max}	3,5	[-]		

Table 9-3: Parameters of the hydrostatic transmission

The blue cells are those which will have the same value for all the different cases, while the yellow cells are those which have to be calculated from the theory or the simulation each time.

The value of the highest process speed Gear Ratio is that one which allows the transmission to change in four seconds the displacement setting of the pump from cero to one without problems of overloading.

With all these parameters in the transmission it has to be obtained the theoretical and the simulated resonance frequency and damping for different working points which are showed in the Table 9-4 and Table 9-5.

ΔP (bar)	U_m	J_t (Kgm^2)	ε_p	ε_m	n_m (rev/min)	ω_h (rad/s)	δ_h
11,217	10	1,100	0,6	1	565,173	**11,459**	**0,297**
14,026	8	1,663	0,6	1	564,704	**9,321**	**0,365**
18,699	6	2,878	0,6	1	563,921	**7,084**	**0,481**
24,932	4,5	5,038	0,6	1	562,879	**5,354**	**0,636**
32,055	3,5	8,263	0,6	1	561,688	**4,181**	**0,815**
12,467	10	1,100	0,6	0,9	627,739	**10,313**	**0,330**
14,024	10	1,100	0,6	0,8	705,882	**9,167**	**0,372**
16,029	10	1,100	0,6	0,7	806,239	**8,021**	**0,425**
18,700	10	1,100	0,6	0,6	939,870	**6,875**	**0,495**
22,441	10	1,100	0,6	0,5	1126,590	**5,729**	**0,595**
15,584	8	1,663	0,6	0,9	627,159	**8,389**	**0,406**
17,532	8	1,663	0,6	0,8	705,146	**7,457**	**0,457**
20,036	8	1,663	0,6	0,7	805,283	**6,524**	**0,522**
23,375	8	1,663	0,6	0,6	938,567	**5,592**	**0,609**
28,048	8	1,663	0,6	0,5	1124,720	**4,660**	**0,731**
20,777	6	2,878	0,6	0,9	626,193	**6,376**	**0,534**
26,716	6	2,878	0,6	0,7	803,686	**4,959**	**0,687**
31,167	6	2,878	0,6	0,6	936,394	**4,251**	**0,801**
37,401	6	2,878	0,6	0,5	1121,590	**3,542**	**0,962**
27,703	4,5	5,038	0,6	0,9	624,906	**4,819**	**0,707**
31,167	4,5	5,038	0,6	0,8	702,295	**4,283**	**0,795**
35,622	4,5	5,038	0,6	0,7	801,562	**3,748**	**0,909**
35,622	3,5	8,263	0,6	0,9	623,437	**3,763**	**0,905**

Table 9-4: Theoretical results for resonance frequency and damping

U_m	ε_m	T (s)	f (Hz)	ω_h (rad/s)		$\omega_h\delta_h$
10	1	0,56	1,786	**11,220**		**3,335**
8	1	0,68	1,471	**9,240**		**3,377**
6	1	0,92	1,087	**6,830**		**3,284**
4,5	1	1,21	0,826	**5,193**		**3,304**
3,5	1	1,60	0,625	**3,927**		**3,200**
10	0,9	0,61	1,639	**10,300**		**3,402**
10	0,8	0,69	1,449	**9,106**		**3,384**
10	0,7	0,80	1,250	**7,854**		**3,335**
10	0,6	0,93	1,075	**6,756**		**3,347**
10	0,5	1,14	0,877	**5,512**		**3,277**
8	0,9	0,77	1,299	**8,160**		**3,314**
8	0,8	0,87	1,149	**7,222**		**3,299**
8	0,7	1,01	0,990	**6,221**		**3,248**
8	0,6	1,17	0,855	**5,370**		**3,271**
8	0,5	1,41	0,709	**4,456**		**3,257**
6	0,9	1,01	0,990	**6,221**		**3,324**
6	0,7	1,33	0,752	**4,724**		**3,245**
6	0,6	1,58	0,633	**3,977**		**3,187**
6	0,5	1,96	0,510	**3,206**		**3,083**
4,5	0,9	1,34	0,746	**4,689**		**3,315**
4,5	0,8	1,57	0,637	**4,002**		**3,183**
4,5	0,7	1,77	0,565	**3,550**		**3,226**
3,5	0,9	1,79	0,559	**3,510**		**3,178**

Table 9-5: Simulation results of frequency

These results shows the values of the damping and the resonance frequency for different values of the gear ratio and the displacement setting of the motor when the displacement setting of the pump makes a step from 0,4 to 0,6 as in the Figure 9-2.

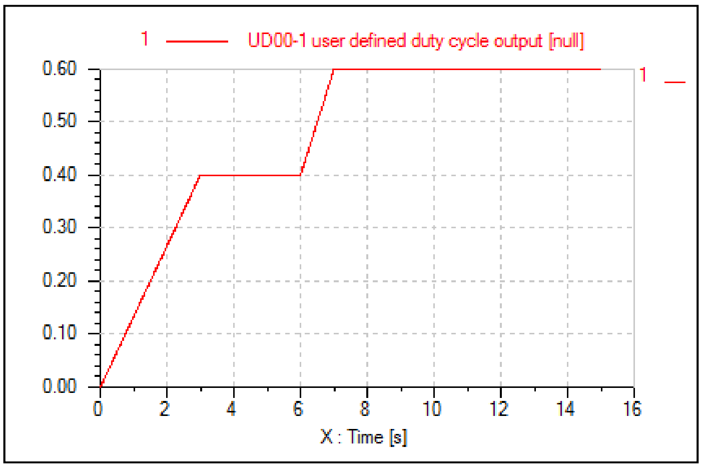

Figure 9-2: Displacement setting of the pump

The step is slow because the system doesn't have any kind of filter to soften the slope when it arrives at the pump.

With those tables can be drawn a graphic which shows one of the most important points of this simulation. First, the range of the resonance frequency (9-15) and damping (9-16), and second the verification of the constant value of $\boldsymbol{\omega_h \delta_h}$.

However, looking at the table can be seen that this is not exactly a constant value. That could be because of the also not constant inertia. To calculate the different points, the gear ratio of the gearbox was changed, so for that reason also the inertia change. Another possible reason could be that the gearbox has its own damping $\boldsymbol{\delta_h}$ and because of that the total damping do not vary linearly with respect to the resonance frequency $\boldsymbol{\omega_h}$ and thus $\boldsymbol{\omega_h \delta_h}$ is not constant.

$$\omega_{hmin} \leq \omega_h \leq \omega_{hmax} \tag{9-16}$$

$$\delta_{hmax} \geq \delta_h \geq \delta_{hmin} \tag{9-17}$$

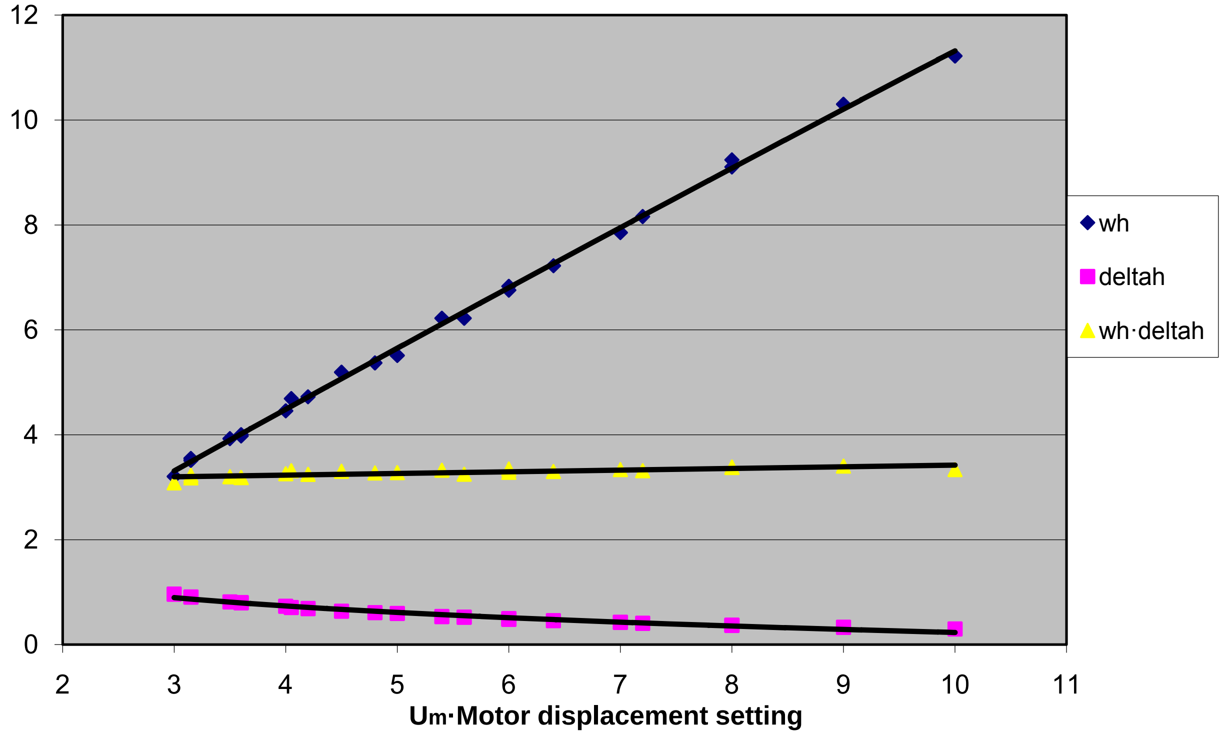

Figure 9-3: Graphic of resonance frequency and displacement setting

All this information is got from the AMESim simulation. Next are going to be shown the high pressure side of three representative cases.

The first one has a big resonance frequency and a low dumping. The third one has low resonance frequency and high damping. The second one is a medium case

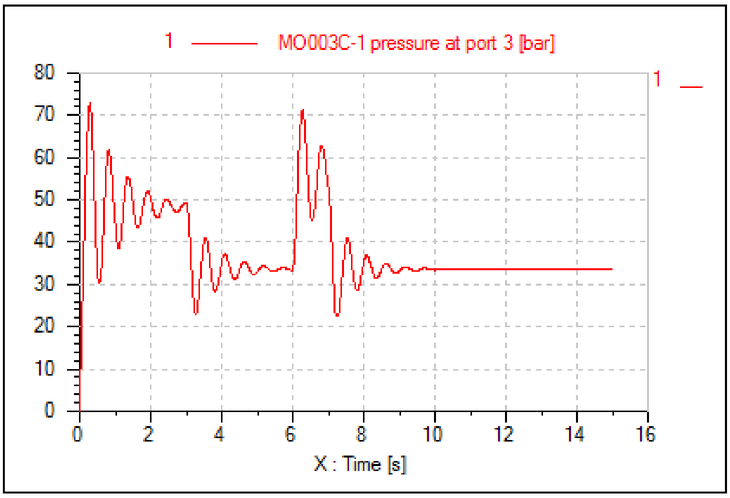

Figure 9-4: High resonance frequency, low damping

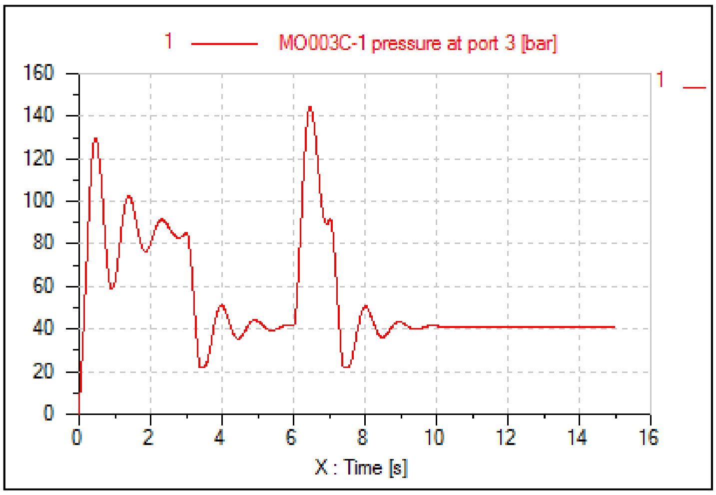

Figure 9-5: Medium case

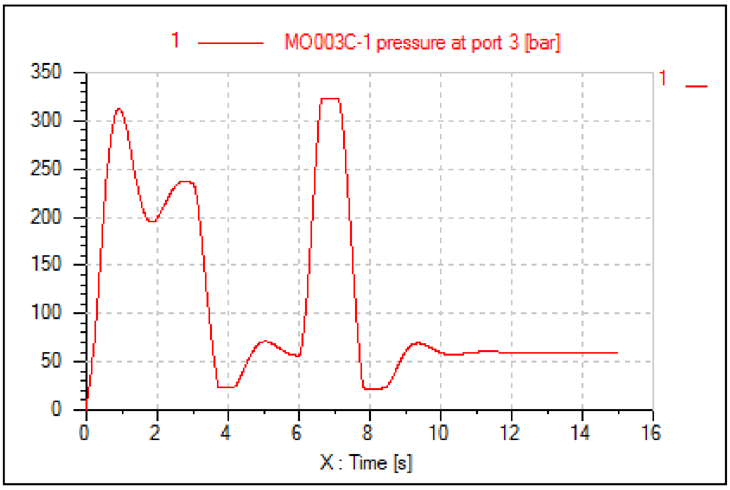

Figure 9-6: Low resonance frequency, high damping

9.3 Angular velocity servo – pump controlled variable motor

9.3.1 Pump displacement setting control

At first it is going to be done an easy feedback where the input is going to be the displacement setting of the pump.

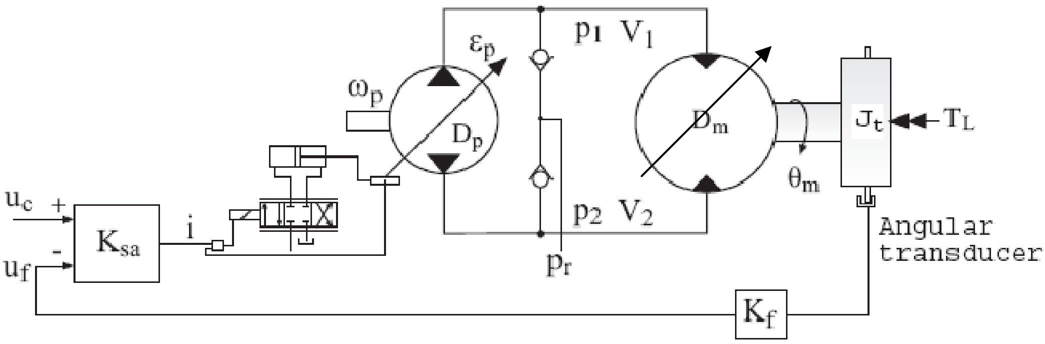

Figure 9-7: Pump controlled motor used as an angular position servo [13]

If the system is approximated to a linearised model the transfer function for the pump displacement controller from input current (i_v) to displacement setting (ε_p) is

$$\frac{\Delta\varepsilon_p}{\Delta i_{ps}} = K_{ps}G_{ps} = K_{ps}\frac{1}{1+\frac{s}{\omega_{ps}}} \quad (9\text{-}18)$$

Combining equation (9-2), (9-3) and (9-18) gives the block diagram shown in Figure 9-8.

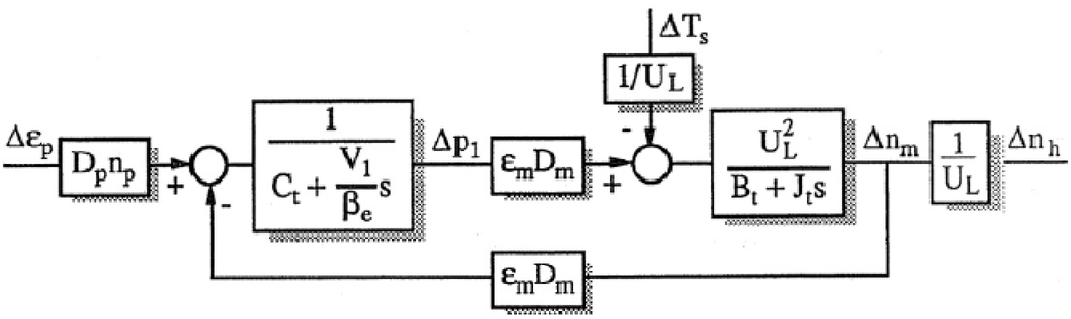

Figure 9-8: Block-diagram for a hydrostatic transmission with gearbox [13]

The AMESim simulation of this first approximation of the feedback is shown in the Figure 9-9.

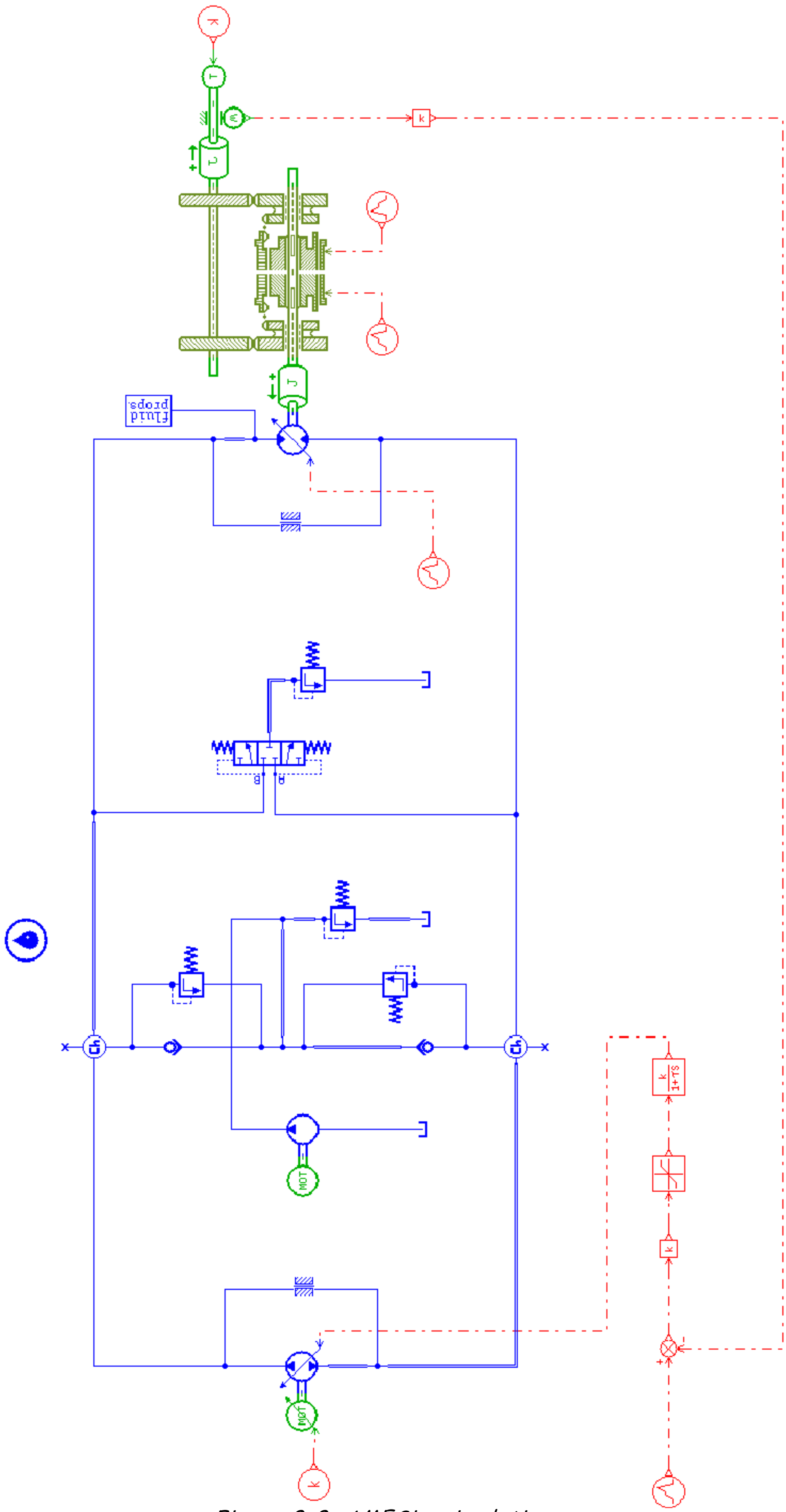

Figure 9-9: AMESim simulation

For this case is done the same study than for the previous one which didn't have feedback and the results are in the Figure 9-10.

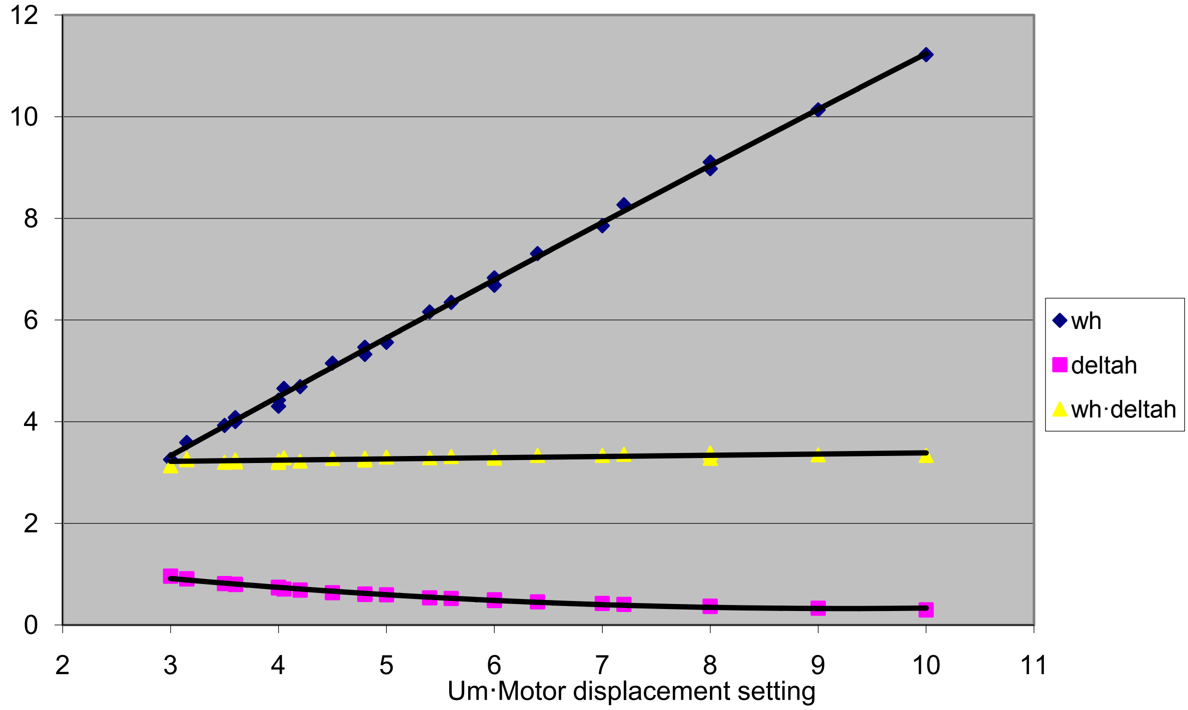

Figure 9-10: Graphic of resonance frequency and displacement setting in feedback system

These results are close to the results of the system without feedback but in this case they are still better because the slope of the previous one is equal to 0,0314 and in the second one (with feedback) is equal to 0,0237.

To understand the effect of the servo system, which the main characteristic is the low pass filter, is going to be shown the input signal before and after it. In the figure 9-11 can be observed that the low pass filter soften the signal.

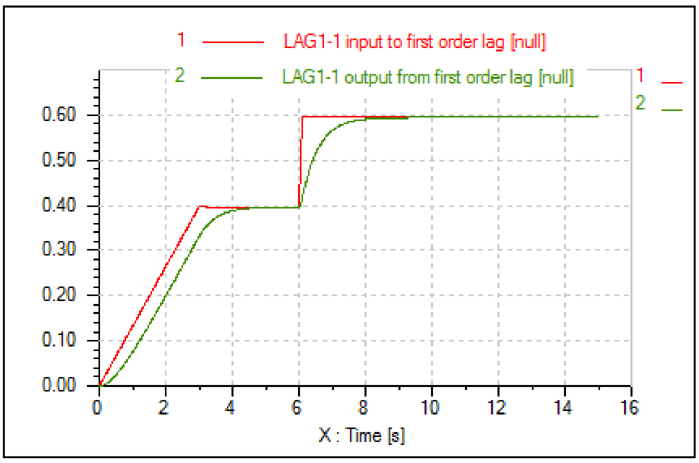

Figure 9-11: Input signal before the low pass filter (red) and after it (green)

9.3.2 Output shaft velocity control

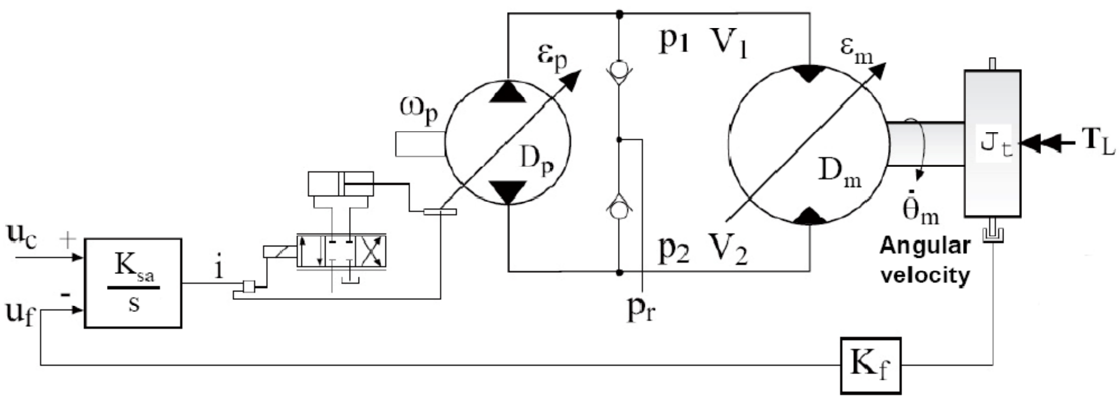

Figure 9-12: Pump controlled variable motor used as an angular velocity servo [13]

In this case the objective is to control from the input the motor velocity. To do it is necessary to add to the feedback a system which can compare the desired velocity with the output velocity and transform it to the necessary pump displacement setting. The system required to do this is a Proportional-Integral-Derivative controller (PID controller). A PID controller attempts to correct the error between a measured system variable and a desired command signal by calculating and then outputting a corrective action that can adjust the process accordingly.

It is also important the low pass filter that has to be before the pump for then soften the input signal and then avoid possible overloading of the transmission or strange behaviours.

In Figure 9-13 is the block diagram of this system. It is important to see the function $\frac{K_{sa}}{s}$ which is the integrator of the PID-controller. This is the function which transforms the velocity into displacement setting.

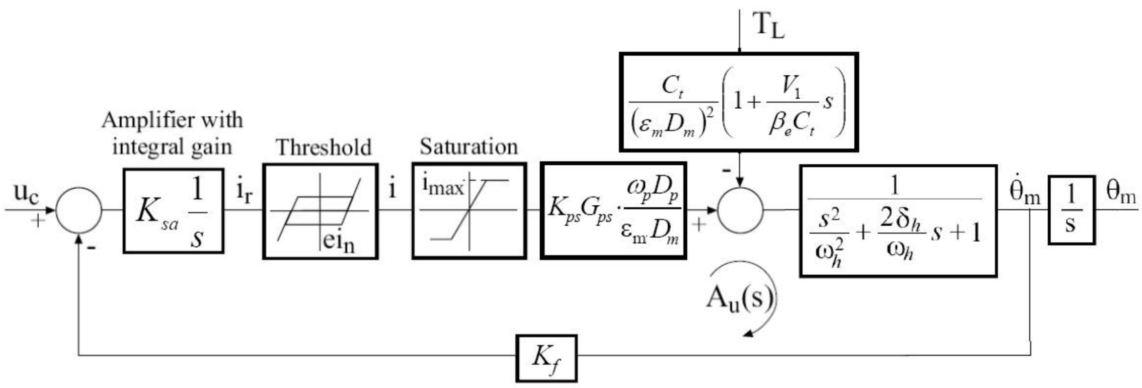

Figure 9-13: Block diagram of a pump controlled motor used as an angular position servo [13]

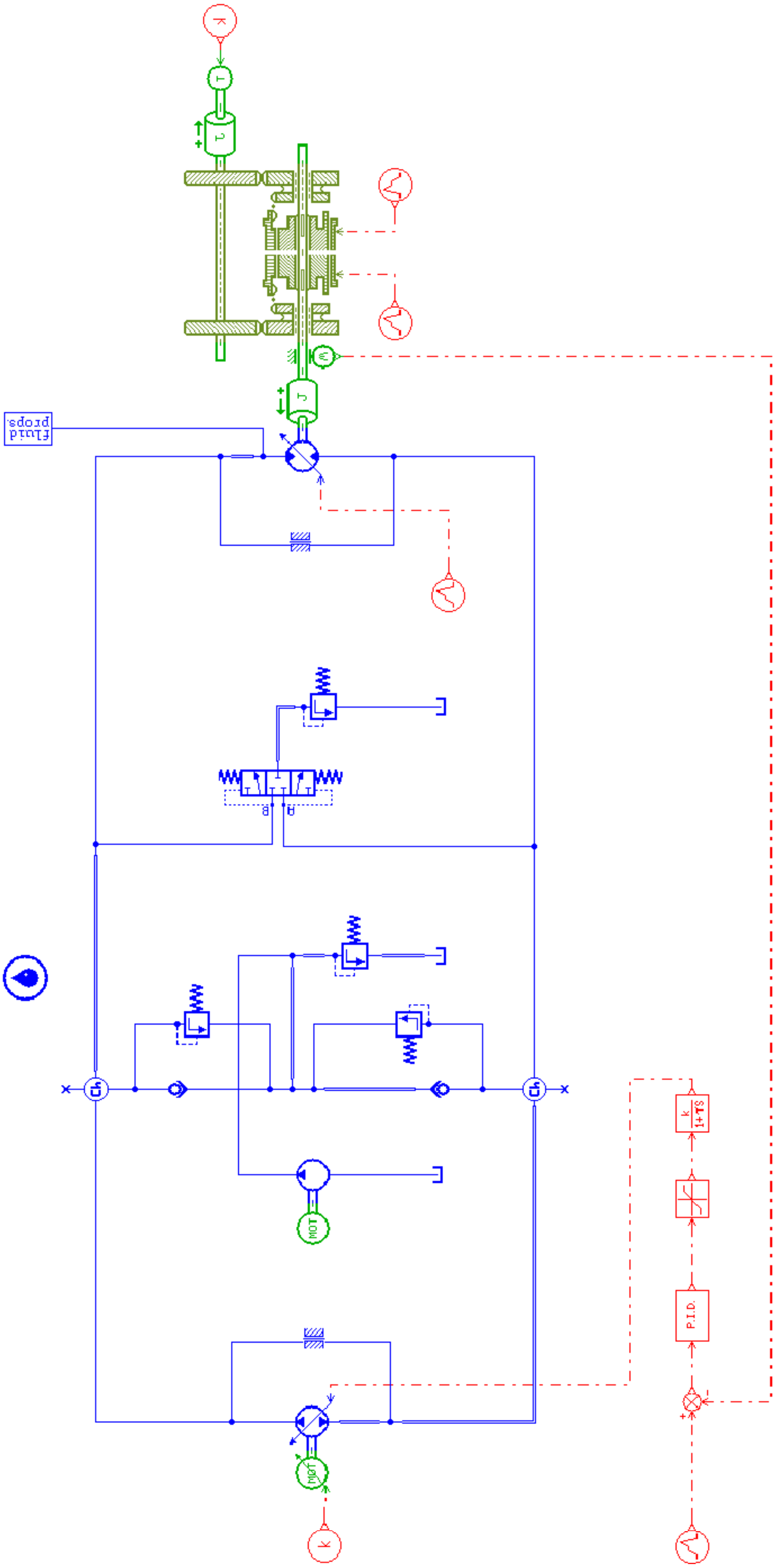

Figure 9-14: AMESim simulation with shaft velocity control

Now are going to be shown some pictures which represent the feedback function beginning for the comparison junction differencing inputs (Figure 9-15), continuing for the PID-controller (Figure 9-16) and finally the low pass filter (Figure 9-17).

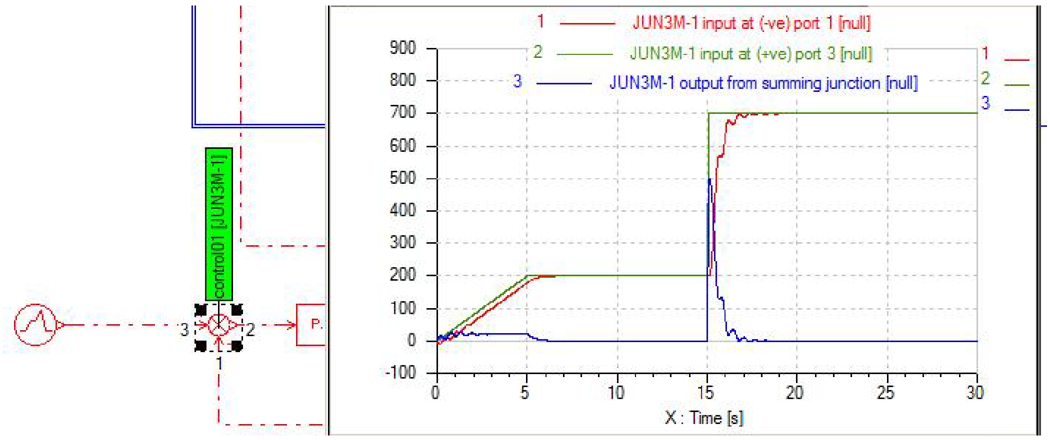

Figure 9-15: Comparison junction

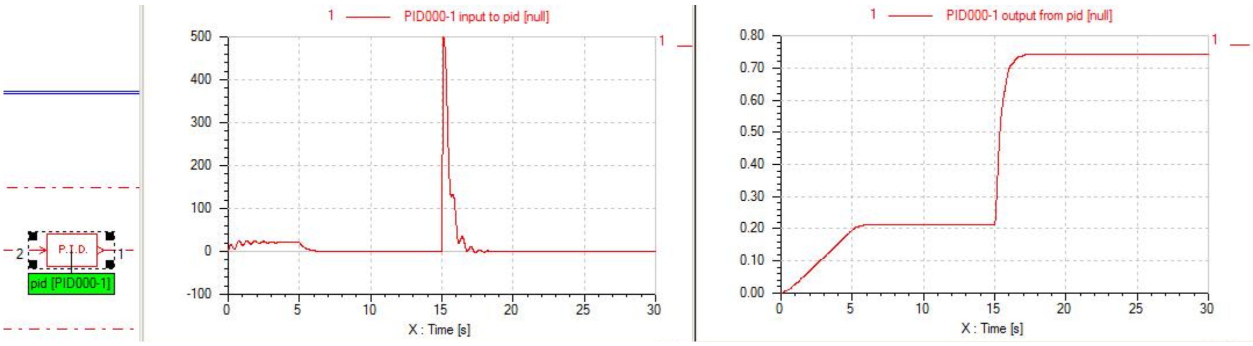

Figure 9-16: PID-controller input (left) and output (right)

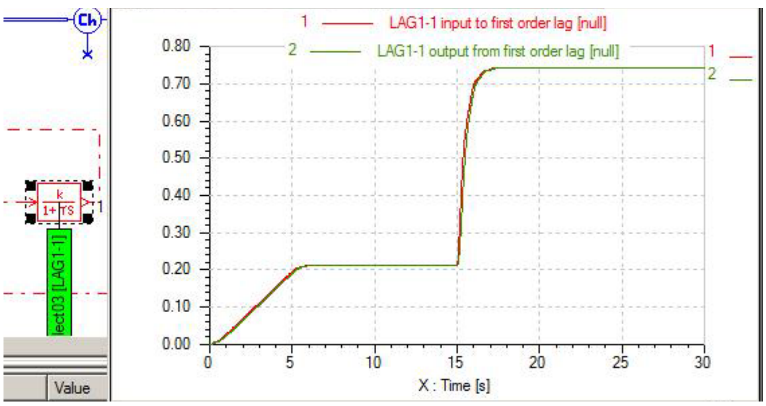

Figure 9-17: Low pass filter

In the Figure 9-15 can be observed that the output signal (blue) is equal to cero from the second 19. This means that the system has reached the desired velocity, so then the output shaft velocity (red line) is equal to the input signal (green line).

It is also important to see in Figure 9-16 that the moment in which the output velocity reach the desired one, the signal turn constant again.

Therefore, as it is shown in the Figure 9-17 the low pass filter doesn't have to do too much because the signal arrives to it very soften.

It is important to analyse the value which has to have the PID-controller gain. It is demonstrated that if it is not changed when the displacement setting of the motor is reduced, in spite of the increasing of the damping the system could be instable.

For example, if the motor displacement setting ε_m =1.0 the control loop is stable. However, for ε_m =0.2 the system is instable, because the loop gain is higher than 1.0 when the phase shift is -180 degrees. (Figure 9-18)

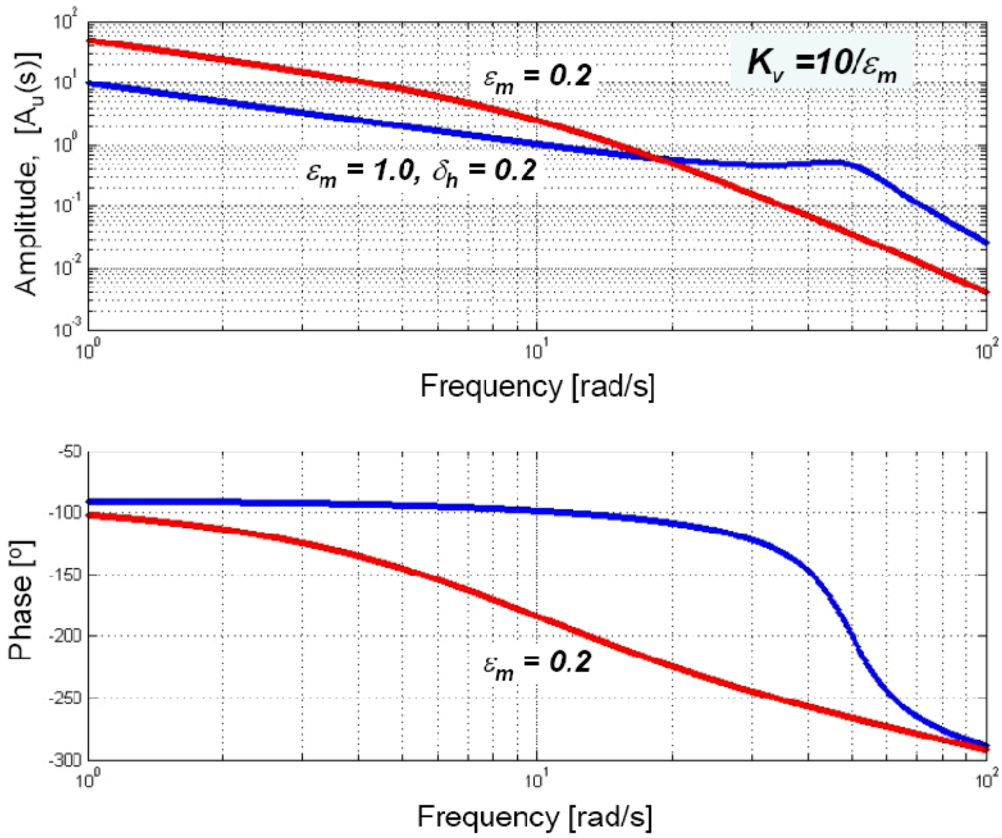

Figure 9-18: Bode-diagram for two different motor displacement settings [13]

The reason for the value of Kv=10/ ε_m is because the open loop gain equation shows that the gain is proportional to 1/ ε_m (9-19)

$$A_u(s) = \frac{K_{sa}K_{ps}D_p\omega_p K_f\frac{1}{\varepsilon_m D_m}}{s\left(1+\frac{s}{\omega_{ps}}\right)\left(\frac{s^2}{\omega_h^2}+\frac{2\delta_h}{\omega_h}s+1\right)} = \frac{K_v}{s\left(1+\frac{s}{\omega_{ps}}\right)\left(\frac{s^2}{\omega_h^2}+\frac{2\delta_h}{\omega_h}s+1\right)} \quad (9\text{-}19)$$

So then a way to solve this problem is to multiply the value of the gain by the displacement setting of the motor. Taking into account the ε_m which is dividing the gain it should be necessary to multiply the gain of the PID-controller by ε_m^2.

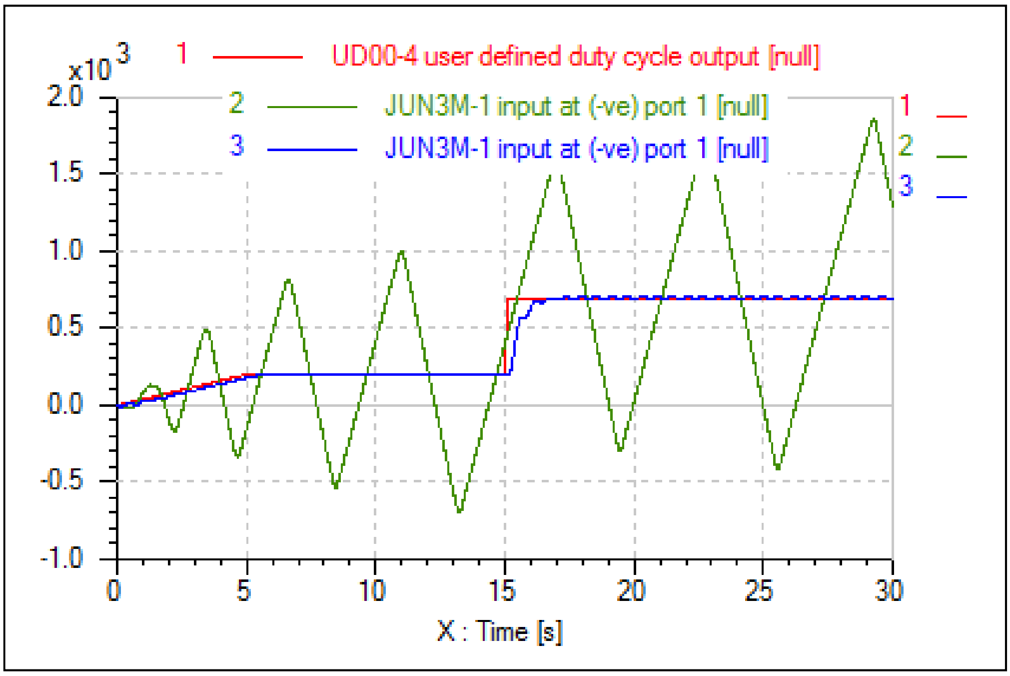

Figure 9-19: Step response when K_v=10/ε_m

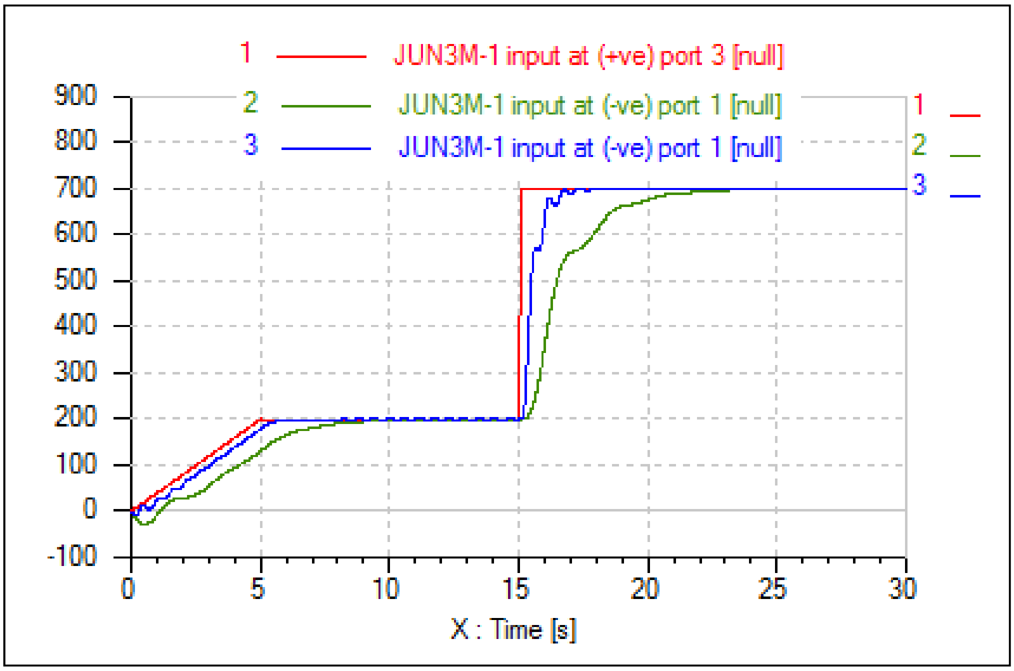

Figure 9-20: Step response when K_v=10·ε_m

In the Figure 9-19 and Figure 9-20 can be observed the input step (red line) the motor shaft velocity when ε_m=1 (blue line) and the motor shaft velocity when ε_m=0,3 (green line). It is very important to see that when the gain is not multiply by the ε_m when the ε_m goes down the system begins to be instable. But when it is multiply this problem is solved.

10. CONCLUSION

In this project have been analysed some different aspects of the hydrostatic transmissions for later, from all that information, design a good hydrostatic transmission to simulate different situations which could be interesting for this kind of systems.

After all, the hydrostatic transmission designed and implemented with AMESim has demonstrated it robust design thanks to the damping system and the safety measures as the cooling system or the overloading valves.

It has been observed that with the laboratory load, which principal characteristic is the Wheaston Bridge, the system has a high damping. Due to this high value the system allows big and quick changes in the input signal without problems of overloading the transmission or long periods with resonance frequency. So the laboratory load of the Department of Fluid and Mechanical Engineering System in Linköping University can support perfect the real analyse of the transmission.

The second load system designed is for a real vehicle application such earth moving machines, or other heavy vehicles but this doesn't have a so big damping. That is the reason for which the transmission is usually overloaded if it has quick changes in the parameters. To solve it the best solution is a low pass filter, which slows down the transmission input changes. Also it is necessary a PID-controller to allow the direct control of the load shaft velocity.

The low pass filter it is a representation of a servo to control the pump. The PID-controller, which is installed to transfer the velocity signal to displacement setting signal, provides a robust and safe control over the whole operating system.

All those control systems have demonstrated that they make a soft system which is most friendly usable.

From other point of view these kind of transmission have demonstrated it high controllability specially in low speed ranges, that is an important reason for it application in heavy vehicles. Also for these vehicles is significant the capacity of the hydrostatic transmissions to operate with high torques allowing power ranges of 200kW.

This one is also the reason for not using them in light vehicles, which necessities are more in higher speeds and it is not so necessary the level of torque that the hydrostatic transmissions are able to support.

So, after the analysis of the obtained results in the different simulations, a suitable combination between the main transmission components, gearbox, and control system, it is possible to reach a high overall efficiency over a wide using range.

Due to the high efficiency levels of the good designed hydrostatic transmissions, it is possible to say that they are interesting from an environment point of view, because the transmission is responsible for most of the fuel vehicles consumption.

Finally the other benefit of the hydrostatic transmissions is that the connection between pump and motor doesn't need any shaft. Due to it the possibilities for location of the different elements are incredible. The wheels can be situated in different levels and it works perfect as can be observed in Figure 11-1. Also they have other special possibilities of location as in a front wheel of a motorbike, to make better traction capacities as in the Figure 11-2, something only possible due to the no shaft necessities.

Figure 11-1: John Deere six-wheel drive heavy machine [20]

Figure 11-2: From wheel hydrostatic transmission in a motorbike [21]

11. NOMENCLATURE

B_m, B_t	viscous friction coefficients	[Nms/rad]
β_e	bulk modulus	[Pa]
C_t	transmission leakage coefficient	[m^5/Pa/s]
D_m, D_p	displacement of motor and pump	[m^3/rad] or [m^3/rev]
δ_h	relative hydraulic damping	[-]
Δp	transmission pressure difference	[Pa]
$\varepsilon_m, \varepsilon_p$	displacement setting of motor and pump	[-]
F_L, F_s	traction force or external disturbance force	[N]
F_L	traction force	[N]
J_t	total load inertia	[kgm^2]
K_f	feedback gain	[V/m] or [Vs/rad]
K_{ps}	pump controller gain	[1/A]
K_{sa}	servo amplifier gain	[A/V]
K_v	steady state loop gain	[1/s]
n_m, n_p	shaft speed of hydraulic motor and pump	[rev/s]
n_h	traction wheel speed	[rev/s]
ω_p	shaft speed of hydraulic pump	[rad/s]
$\dot{\theta}_m = \omega_m$	shaft speed of hydraulic motor	[rad/s]
P_e, P_{in}	engine or input power	[W]
T_L	external disturbance torque	[Nm]
U_m, U_L	mechanical gear ratio	[-]
V_1	pressurized transmission volume	[m^3]
ω_h	hydraulic natural frequency	[rad/s]
ω_{ps}	pump controller bandwidth	[rad/s]
q	flow	[m^3/s]
η_v	volumetric efficiency	[-]
η_{hm}	hydraulic mechanical efficiency	[-]
p_1, p_2	pressure values	[Pa]

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